GPO PRICE $ ______

CFSTI PRICE(S) $ ______

Hard copy (HC) $3.00

Microfiche (MF) .75

ff 653 July 65

N66 25571 (ACCESSION NUMBER)

(THRU)

43 (PAGES)

1 (CODE)

CR-65340 (NASA CR OR TMX OR AD NUMBER)

15 (CATEGORY)

FACILITY FORM 602

THE Marquardt CORPORATION

PROPRIETARY INFORMATION

This document contains information proprietary to The Marquardt Corporation, subject to any rights vested in the Government by contract, and may not be used, reproduced, published, or distributed in any form or disclosed in whole or in part without the authorization of The Marquardt Corporation except by the Government pursuant to its rights. All use by, or with the approval of, the Government, is authorized.

3 JANUARY 1965 REPORT NO. S-479

FINAL REPORT

TWO (2) WAY, LATCHING, DC SOLENOID CONTROL VALVE

During the Period
1 March to 28 December 1965

NASA Contract NAS9 - 4069

TMC Project No. 3002

PREPARED BY:

R. A. Lynch
R. A. LYNCH

APPROVED BY:

H. Wichmann
H. WICHMANN

E. P. Perlman
E. P. PERLMAN
Program Manager

CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Continued)

LIST OF TABLES

I. INTRODUCTION

On 1 March 1965 The Marquardt Corporation was awarded contract NAS 9-4069 by the NASA Manned Spacecraft Center in Houston, Texas. This contract was for a six month development program to design, fabricate, develop and deliver to NASA, three two (2) way, latching, D.C. solenoid control valves suitable for use in hypergolic propellant systems. The intended application of the valves is to isolate reaction control engine clusters from main propellant supply systems in space vehicles.

The intent of the program was to design and develop a valve design incorporating the Marquardt bistable actuator, which is ideally capable of meeting the performance and environmental conditions typical of the Apollo Spacecraft reaction control system. The primary design objective was to achieve the required performance characteristics with the minimum possible unit weight. The work in support of this objective was completed on 28 December 1965.

This report describes the effort in support of the subject contract. It describes the analytical, design and development test activities and presents performance data for the developed configuration.

II. SUMMARY

During the period from 1 March to 28 December 1965 The Marquardt Corporation performed the tasks required under Contract NAS 9-4069. The objective of this program was to design and develop a two way, latching D.C. solenoid valve for use in propellant systems. A photograph of this valve is shown in Figure 1. The contract also provided for the fabrication of three units, of the developed design, for delivery to NASA.

The three units fabricated for delivery incorporate the features evolved as a result of prototype testing. These units also incorporate all welded construction to assure minimum external leakage and a fully hermetic seal at the electrical pigtail interface.

Though not required by the scope of this contract, the units fabricated for delivery were tested for compliance with the performance requirements of the design specification (Appendix A). A summary of demonstrated performance follows.

Criteria	Specification Requirement	Demonstrated
Internal helium leakage at 360 psig	20 scc/hr max	0
ΔP at equivalent flow rate of 0.88 pps N_2O_4	2.0 psi max	2.55 to 2.90
Insulation resistance	500 megohms min	> 500 megohms
Unit weight	1.5 lbs	1.565 lbs

Unit weight can be reduced to approximately 1.30 lbs by more rigorous contouring of the assembly and appropriate selection of a pigtail. Pressure drop can be reduced by improving flow path area transitions more consistent with the prototype unit. In addition to the above criteria, performance of the position indicator was marginal and a replacement of the magnetic reed switches with miniature microswitches is proposed to rectify this characteristic.

The design concept selected, incorporated the Marquardt bistable actuator concept, welded metal bellows dynamic seals, and a Teflon poppet-seat interface seal. A prototype unit was fabricated to evaluate the selected concepts and tests performed to demonstrate capability and design margins. The prototype unit differed from the final design only in the boilerplate concept utilized to facilitate simple and rapid modifications during the development phase. The prototype unit demonstrated satisfactory performance and design margins after a minimum of modifications. Development testing yielded significant data on the performance of welded metal bellows in pressurized fluid systems and also demonstrated the need for still further information to facilitate the design of bellows into fluid systems.

III. DISCUSSION

A. Design Analysis

The following analyses develop design criteria for a valve meeting the requirements of the specification presented in Appendix A.

1. Force Analysis

To achieve the required flow and pressure drop characteristic, with minimum unit weight, a fully balanced poppet design was selected. From prior valve experience, an equivalent orifice coefficient of 0.65 is used to conservatively estimate stroke requirements.

Requirements

$$\dot{W}_o = \text{Flow rate} = 0.88 \text{ lb/sec } N_2O_4$$

$$\Delta P \text{ at } \dot{W}_o = 2.0 \text{ psi}$$

For liquid flow through orifices

$$Q = 29.81\ C_d\ d^2\ \sqrt{\Delta P/\text{sp. gr.}}$$

Where

Q	=	Flow rate in gpm
C_d	=	Orifice coefficient = 0.65 for assumed sharp edge orifice
d	=	Diameter in inches of equivalent orifice
ΔP	=	Pressure drop in psi
sp. gr.	=	Specific gravity of fluid

$$\text{sp. gr. } N_2O_4 = 1.49$$

$$\dot{W}_o = 0.88 \text{ lb/sec} \times 60 \text{ sec/min} \times \frac{1 \text{ gal}}{11.94 \text{ lb}} = 4.43 \text{ gpm}$$

$$4.43 = 29.81 \times 0.65\, d^2 \sqrt{\frac{2.0}{1.49}}$$

$$d^2 = 0.1975$$

$$\text{Flow area} = \frac{\pi}{4} d^2 = \frac{\pi}{4} \times 0.1975 = 0.155 \text{ in.}^2$$

Disregarding the seat angle affect, and assuming the minimum flow area occurs at the seating diameter,

$$\text{Flow area} = \pi d_1 \times S$$

Where

d_1 = Valve seat diameter in inches

S = Effective valve poppet stroke in inches

Therefore, for design analysis

$$\pi d_1 S = 0.155$$

The selection of a valve seat diameter shall be limited to those diameters for which metal bellows are readily available. From these values the following table is developed.

TABLE I

VALVE SEAT DIAMETER VS. STROKE

d_1 (inches)	S (inches)
0.562	0.088
0.687	0.072
0.812	0.061
0.937	0.053
1.062	0.047

A second parameter influencing selection of seat diameter, is the required poppet force to achieve the specified leakage requirement (20 scc/hr of helium at 360 psig). From prior experience with soft seat (Teflon) configurations, this seal can be accomplished with a seat loading of four pounds per linear inch of seat circumference. The values of Table II are therefore derived from $F = 4 \pi d_1$.

TABLE II

VALVE SEAT DIAMETER VS. REQUIRED SEAT LOAD

d_1 (inches)	F* (lbs)
0.562	7.05
0.687	8.65
0.812	10.20
0.937	11.80
1.062	13.25

* F = Seat load required

The characteristics of Tables I and II are plotted in Figure 2 and a seat diameter of 0.812 inch was selected. The balance of the design analysis will allow for increased poppet stroke to assure meeting the pressure drop requirement.

Valve Closed Force Requirements

In addition to the load requirements to achieve the seal at the poppet seat interface, the closing force must also be selected to assure sealing under vibration, acceleration and shock environments, and any unbalance of the poppet and bellows. For environmental loads, a worst case of 20 g is selected and an effective poppet mass of 0.20 lb is estimated.

$$F_g = 20 \times 0.2 = 4.0 \text{ lbs}$$

From characteristic curves of metal bellows the variation in effective area over the pressure range is typically within $\pm$ 1% of the nominal effective area. For worst case

$$F_{T_c} = 10.20 + 4.0 + \frac{\pi}{4} (0.812)^2 \times 360 \times (0.01)$$

$$F_{T_c} = \text{Total closed force required}$$

$$= 14.20 + 1.87 = 16.07 \text{ lb.}$$

Valve Opening Force Requirements

Since the bistable concept negates the latch force effects during actuation, the actuator must only provide sufficient pull force, at the specified valve stroke, to open the valve against worst case poppet force conditions. Contributing factors are: g load due to vibration, acceleration and shock, unbalance due to area differentials, and any preload in the bellows. For the purpose of initial design, the actuator is assumed to be a bistable actuator with latching forces of 15 pounds in both open and closed positions.

$$F_{opening} = 20 \text{ g } (0.2 \text{ lb}) + \frac{\pi}{4} (0.812)^2 \times 360 \ (0.01) + 1.5 \text{ lbs}$$

$$= 4.0 + 1.87 + 1.5 = 7.37 \text{ lbs minimum}$$

Valve Opened Force Requirements

As the valve opens the spring rate of the metal bellows creates additional force requirements. Since the opened latch force has been selected as 15 pounds and the valve stroke as 0.065 inch, the maximum allowable bellows spring rate can now be derived.

$$F_{T_O} = 15.0 = 20\ g\ (0.2\ \text{lb}) + \frac{\pi}{4}\,(0.812)^2 \times 360\ (0.01)$$

$$+ 1.5 + K_{\text{bellows}}\ (0.065)$$

$$0.065\ K_{\text{bellows}} = 15.0 - 7.37 = 7.63\ \text{lbs}$$

$$K_{\text{bellows}} = \frac{7.37}{0.065} = 117.4\ \text{lb/sec}$$

Since two bellows will be acting in series, their spring rates will be additive. Assuming both bellows are identical, the spring rate of each must not exceed 117.4/2 = 58.7 lb/in.

Valve Closing Force Requirements

Once the opened latch force is overcome, the closing force need overcome only that portion of the worst case environment forces which are not overcome by the bellows spring load.

$$F_{\text{closing}} = 20\ g\ (0.2\ \text{lb}) + \frac{\pi}{4}\,(0.812)^2\ 360\ (0.01) - (1.5 + 7.63)$$

$$= 5.87 - 9.13 = -\ 3.26\ \text{lbs}$$

Therefore, no closing pull force is required and the closing coil of the actuator need only be sized to reduce the opened latch force so that the difference between open and closed latch force is 3.26 lbs.

Valve and Actuator Design Force Criteria

Actuator closed latch force	=	15.0 lbs
Actuator opened force	=	15.0 lbs
Actuator opening force at 0.065 stroke	=	7.37 lbs
Actuator closing force at 0.065 stroke	=	0 lbs
Bellows closing preload	=	1.5 lb
Bellows maximum spring rate	=	58.7 lb/in.

2. Poppet-Seat Interface

Several factors must be considered to optimize the poppet-seat configuration. Of prime concern is the establishment of a stable, repeatable seat diameter. Other considerations are smooth flow transitions across the valve seat with the valve open, repeatable valve strokes and soft seat compression, and efficient utilization of actuator forces in the closing mode. The selected poppet-seat interface employs a 90° included angle hard seat in the valve body and a formed Teflon soft seal carried on the poppet. This design was selected on the basis of the critical finishing and inspection of the Teflon being greatly simplified since it is readily accessible. Other inherent advantages are realized since increased closing force tends to reduce the effective seat diameter while inlet pressure tends to increase the seat diameter. Therefore the seal can be designed to give optimum performance at the most critical pressure conditions. The formed Teflon soft seal configuration was selected to allow maximum Teflon volume for distribution of seating stresses and assure resilience of the seal. The poppet to seat differential angle of 5° which was selected to control Teflon compression, results in a lip seal configuration of the Teflon on compression without protrusion of a thin Teflon section into the flow stream when the valve is open, and provide a well defined metal to metal land on the upstream side of the interface to control Teflon compression and prevent extrusion of the soft material. The installation of the Teflon seal into the poppet is accomplished with a minimum of special tooling and the finished surface is readily inspected without requiring inspection equipment to contact the critical surface.

3. Poppet Guidance

The ball joint shaft linkage between poppet and actuator armature, though more complex than a rigid one piece poppet and shaft with mechanical linkage to the armature, was based on sound analysis. To assure repeated seating of the poppet it is imperative that the poppet be accurately constrained. By machining the guide bore integral with the body seat and the guide diameter integral with the seal defining portion of the poppet, eccentricity and squareness errors are minimized. The location of the guide bore as close to the seat as possible also reduces the moment arm to the seating diameter to minimize misalignment due to the guide clearance. The metal bellows, which inherently impart side loads, straddle the guide bore and their effects on poppet alignment are minimized.

The ball joint shaft linkage, ball diameter and ball location were selected to assure that forces applied to the poppet from the actuator armature would always create a righting moment should the poppet cock sufficiently to contact the guide bore. The shaft is therefore effectively pulled through the close clearance bore rather than pushed. This joint also precludes the requirement of close alignment control between actuator and valve sections, and side loads, inherent in magnetic devices are not imparted to the poppet.

4. Bellows Configuration Selection

On completion of the preliminary design analysis, a preliminary specification for the metal bellows was prepared and submitted to three manufacturers of welded metal bellows (Sealol, Inc., Metal Bellows Corp., and Bell-Metrics Corp.). A welded bellows was selected because of its superior flexibility, pressure resistance and life and the greater selection of compatible materials available. For purposes of the initial quotation, the material was limited to AM350 because of its documented compatibility with the operational fluids and desirable characteristics when fabricated into welded bellows. Sufficient information was not available to firmly establish the best bellows plate configuration (flat plate, nesting ripple, single sweep, torus) and vendors were instructed to verify the basis of their plate configuration selection.

All vendors responded and the flat plate configuration proposed by Bell-Metrics Corp. was selected. Test data on similar bellows, operating in the 0 to 500 psi pressure range indicated a stable effective area over the range. The simple configuration of the plate also assures consistent effective areas between convolutions and from bellows assembly to bellows assembly. These advantages are achieved at some sacrifice in flexibility and pitch (spacing between convolutions) but the overall performance was expected to warrant this selection.

5. Actuator Design Analysis

Utilizing the force requirements derived and the electrical requirements of the contract work statement the analytical design of the actuator was initiated. The selection of "MARQ-METAL" for the magnetic core material was based upon its demonstrated superior properties in previous TMC valve development work and the dry solenoid valve design.

The cross sectional area of the working gap was determined by estimating the average path length, total air gaps and a flux density. An Alnico V permanent magnet was selected on the basis of availability in ring magnet form, well documented characteristics and the negligible weight savings achieved by using other magnetic materials. Coils were then sized to provide sufficient flux to shift the permanent magnet flux and provide adequate pull force at the design stroke. This analysis was repeated a number of times at various assumed flux densities at the air gap to establish an optimum relationship between working gap area, coil size and flux leak paths.

The optimum configuration was established at an air gap flux density of 7800 gauss.

$$B = \frac{1.73\ F}{A}$$

Where

B = Flux density = 7.8 k gauss

F = Force = 15 lbs

A = Working gap area - $in.^2$

$$A = \frac{1.73\ F}{B^2} = \frac{1.73 \times 15}{(7.8)^2} = 0.43\ in.^2$$

The permanent magnet dimensions were then derived to provide sufficient magnetizing force to generate the required flux density at the working area and throughout the magnetic path.

$$NI = 2.02\ B \times S$$

Where

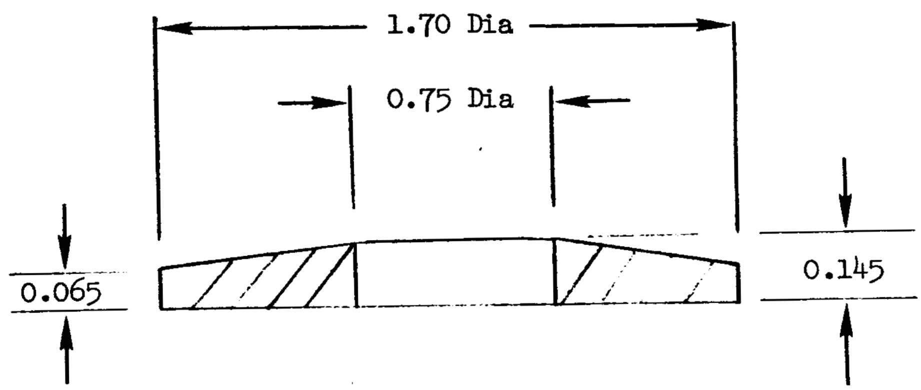
NI = Magnetizing force - amp turns

B = Flux density - gauss

S = Stroke or gap - in.

$$NI = (7800 \times 0.009 + 2.0 \times 10)\ 2.02 = 183\ AT$$

Allowing for 15% flux leakage, 210 AT magnetizing force is required. Using Alnico V permanent magnet material, a ring magnet of the dimensions shown below is required.



Flux density to cause the maximum required opening force was then calculated from the relationship:

$$B = \frac{1.73 \text{ X } F}{0.43}$$

$$B = \frac{1.73 \times 7.65}{0.43} = 5550 \text{ gauss}$$

From previous bistable actuator development, additional flux must be generated by the coil to effect the shifting of the permanent magnet flux from the closing to the opening coil loop. The opening coil was therefore sized to generate a flux of 6700 gauss. The magnetizing force was calculated from the relationship

$$NI = 2.02 \text{ B} \cdot \text{S}$$

$$NI = 2.02 \times 6700 \times 0.074 + 2.02 \times 2 \times 10 = 1040 \text{ A - T}$$

Allowing 20% leakage flux the coil was sized to generate 1230 amp turns of magnetizing force.

From worst case temperature conditions and to allow some margin the minimum operating current of the coil was established as 1.2 amp.

$$\text{No. turns required} = \frac{NI}{\text{Current}} = \frac{1230}{1.2} = 1030 \text{ ampere - turns}$$

The coil diameters (ID and OD) were then established to be consistent with the permanent magnet diameters to simplify the assembly and optimize the unit weight. The average turn length of 3.6 inches/turn required a total length of coil wire of 312 feet. For a coil resistance of 15 ohms at 20°C, the selected coil wire requirement is 15 ohms/312 feet = 48 ohms/1000 feet. The coil may therefore be wound of #24 aluminum wire or #26 copper wire.

To effect closing, the closing coil need only shift sufficient flux from the opening coil loop to the closing coil loop to negate the latch force. Considering a 25% flux leakage approximately 5000 gauss need be generated by the closing coil. The magnetizing force to generate this flux density was calculated from the relationship:

$$NI = 2.02 \, B \cdot S = 2.02 \times 5000 \times 0.074 \approx 800 \text{ ampere - turns}$$

For a mean turn length of 3.6 inches and an operating current of 1.2 amps the coil may be wound of 670 turns of #26 aluminum wire or #28 copper wire.

With reference to the B - H characteristic of the selected core material, the cross sectional areas and dimensions of the core path were determined at a flux density of 20,000 gauss.

$$\text{Total required flux} = 7800 \text{ gauss} \times 0.43 \text{ in.}^2 = 3350 \text{ gauss in.}^2$$

$$A_{\text{core path}} = \frac{3350}{20{,}000} = 0.168 \text{ in.}^2$$

The actuator weight was then estimated for aluminum coils and copper coils. The aluminum coil design was selected on the basis of its lighter weight.

6. Position Indicator Selection

Two methods of providing an electrical signal to monitor valve position were investigated. A microswitch, located so as to be actuated by an extension of the valve shaft, was selected and mounting concepts studied. The designs all required additional valve assembly components to assure rigid and positive positioning and an adjustable valve shaft extension. The switches selected required added unit height as they could not be readily contained with within the actuator envelope.

Magnetic reed switches, such as those manufactured by General Electric and Hamlin Inc., were investigated. A single pole double throw subminiature magnetic reed switch manufactured by Hamlin Inc., was selected. The size of these units would allow the installation of two switches in parallel within the actuator envelope with no increase in unit height. The parallel circuit would provide a redundant indicator circuit and also assure a minimum apparent circuit contact resistance. Two methods of actuation were proposed. With a small permanent magnet mounted on the valve shaft, and the reed switches mounted perpendicular to the actuator axis so as to be isolated from the actuator induced field, the switch would respond to the proximity of the shaft mounted permanent magnet thus monitoring valve position. The marginal aspects of this scheme, which could only be verified by test were:

(1) The ability to isolate the reed switch from actuator induced fields

(2) The sensitivity and hysteresis characteristics of the reed switch as to its ability to discriminate a difference in permanent magnet position of approximately 0.065 inch.

The second, and simplest method, was to mount the reed switch in such a position and orientation as to discriminate between leakage fluxes with the armature in the closed position and leakage fluxes in the open position. This method was selected as the primary technique to be evaluated during development testing. The switch was to be located parallel to the actuator axis and within the counter bore existing in the pole piece. The exact location and orientation was to be determined by test.

7. Material Selection

Prime consideration for materials used in wetted portions of the valve was compatibility with the working fluids, test fluids and combinations of these. Type 304 and 321 exhibited the most desirable properties but do not have a heat treat capability. Those parts, except bellows and actuator core, which required heat treat to obtain more desirable hardness and wear resistance were to be fabricated of 17-7PH precipitation hardenable corrosion resistant steel. In areas expected to experience wear or sliding contact with mating components, one surface was "Electrolized" to improve wear resistance and also provide lubricity.

B. Development

1. Prototype Unit Design

Concurrent with the design analysis, the design of the prototype unit was initiated. The primary objective of the design was to simulate, as much as possible, the internal configuration of the production unit. However, the design must also provide for modifications to the flow path to improve the Δ ρ characteristic, allow for adjustments of bellows position and poppet stroke, and allow access for measuring valve and actuator component characteristics. The valve section and actuator section were designed as separate assemblies with test fixtures to simulate the interfaces. The capability to ultimately assemble the actuator to the valve section was also retained. The wetted portions of the valve section, duplicated in most every detail, the planned production unit configuration while bolted construction and the use of elastomeric static seals permitted disassembly and assembly, and ease of adjustment and measurement of valve stroke and force. The materials of construction of the prototype were identical with the planned production unit design so as to evaluate characteristics of sliding fits and the poppet metal-to-metal stop.

2. Prototype Unit Fabrication

The fabrication of the prototype unit was accomplished at Marquardt. Most parts were made on lathes and mills without the need for special tooling. Critical sliding surfaces and contacting surfaces were finish-ground after heat treat and critical wear areas were electrolized. The prototype unit is shown in Figures 3, 4, 5, and 6.

Actuator coils were wound of aluminum wire on nylon spools. Teflon coated copper leads were then soldered to the aluminum wire with a 90% tin-10% zinc solder using an ultrasonic soldering pot. The copper leads were then wound one turn each in the proper direction and brought through the access slots in the spool. The coil was then finished by wrapping the wire bundle with mylar tape.

Assembly of the poppet assembly was accomplished by forming the flat Teflon seal to the poppet contour at 200°F with a dummy retainer. With the dummy retainer clamped in place the assembly was cooled in liquid nitrogen for five minutes. The dummy retainer was then removed and the retainer, heated to 200°F, pressed in place. The Teflon was then cut back to the specified protrusion on a lathe after the assembly had been stabilized by heating to 300°F for 8 to 10 hours and air cooling to ambient temperature for no less than 12 hours. The finished poppet assembly was then deburred under a microscope and the Teflon examined microscopically for surface condition and a profile traced on a comparator.

The valve section and actuator sections were assembled separately and submitted for development testing.

3. Prototype Testing

a. Actuator Section

After checking out the prototype actuator for continuity, coil resistances and shorts, the actuator was polarized by applying 32 Vdc to both coils while the coils were connected in parallel. Armature stroke was measured with a dial indicator and the force characteristics of the assembly determined. Dead weights were suspended from the armature shaft and various input voltages applied. Pole face gaps in the latched position were varied by the position of an adjusting screw on the mounting plate. Force, voltage and pole face gap characteristics of the prototype actuator are shown in Figures 7 and 8.

During testing it was evident that force outputs of the actuator were below the design levels. The assembly was therefore repolarized using an R.F.L. Magnet Charger. A 400 Vdc pulse was applied to the parallel connected coils. Some increase in force output was noted as a result of the stronger polarizing force, but design levels were not achieved. From the force vs. coil voltage characteristics obtained, it was determined that the magnetic circuit was saturating at a low input level. The magnetic circuit design was carefully reviewed and some cross sectional areas found to be marginal. A new armature was fabricated and the marginal cross sectional areas of the housing and pole piece were increased, to obtain a consistent path cross-sectional area, by press-fitting rings of the core material. On retest, the actuator force and stroke capability exceeded the design levels by a small but adequate margin (Refer to Figures 9 and 10).

The actuator was then instrumented to determine dynamic electrical characteristics. Induced voltage traces were obtained to establish need for voltage spike suppression. Induced voltage spikes of 50 to 100 V were measured under various conditions of applied voltages and armature force output. A zener diode circuit to limit the magnitude of these spikes to less than 40 volts was designed for incorporation in the production design.

A magnetic reed switch was then wired to a battery and indicator light and mounted in the pole piece counter-bore. The actuator was subjected to various opening and closing voltages with the switch in various orientation. With the switch located in contact with the pole piece counterbore wall and with at least 2/3 of its length inserted into the bore, positive monitoring of armature position was obtained. In this position the switch was insensitive to coil energization and only responded to armature position. This scheme was therefore adopted for incorporation into the production design and the other schemes were not tested. Typical actuator test set ups are shown in Figures 11 and 12.

b. Valve Section

After checkout of the flow and pressurization system set up, the valve section was installed for the flow vs. pressure drop test at various poppet strokes. The valve stroke was set by a mounting plate adjusting screw and measured by a dial indicator. Initial characteristics indicated a fixed restriction existed which prevented reduction in pressure drop at strokes above 0.060 inch. It was determined that the lower bellows, on poppet stroking, extended to restrict the flow passage at strokes above 0.060 inch, the increased restriction due to bellows extension into the flow stream was equal to the reduction in restriction due to the poppet to seat clearance. Spacers and a new adapter ring were fabricated to relocate the lower bellows out of the flow stream and flow vs. $\Delta\rho$ tests performed. At the design stroke of 0.065 inch a minimum $\Delta\rho$ of 2.7 psi was obtained at a flow rate of 0.73 pps of water (equivalent to 0.88 lb/sec of N_2O_4). By more stringent contouring of the flow passages downstream of the seat a $\Delta\rho$ of 2.0 psi was obtained at the rated flow condition. In evolving this ultimate pressure drop, the effects of contouring the inlet passage and/or shrouding the inlet bellows to prevent direct fluid impingement, were tested and found to be negligible. Pressure drop with reverse flow was also measured and found to be 10 to 20% higher than forward flow at the same stroke and flow rate.

Concurrent with the flow vs. $\Delta\rho$ tests, force vs. deflection characteristics of the poppet with various system pressures, were determined. For strokes up to the design stroke of 0.065 inch the gross spring rate was approximately 100 lb/in. up to system pressures of 100 psig. At higher pressures, a radical increase in spring rate occurred (See Figures 13 and 14). A comparison of this characteristic with a force vs. pressure at zero deflection, supplied by the bellows manufacturer, indicated that the pressures above 100 psi were resulting in a deformation of the flat plate convolution configuration sufficient to cause contact between adjacent plates thereby altering the spring rate characteristic. To verify this assumption and determine corrective action, the bellows assembly solid height was measured. The bellows were then assembled into the valve at various installed heights, such that no seat preload existed to assure that both bellows were at equal heights. The installed dimension yielded a value of available stroke per convolution at installation by the following relationship:

$$\frac{h_{installed} - h_{solid}}{N}$$

Where

$h_{installed}$ = Bellows height at installation

h_{solid} = Bellows solid height

N = Total number of convolutions

Force-stroke tests at various pressures were then performed for each installed height and spring rate curves generated for each installed height. From these plots, the pressure at which the spring rate exceeded 117 lb/in. (the design limit for the two bellows in series) was determined and plotted against the available stroke per convolution at installation (Refer to Figure 15). From the plot thus obtained it was determined that an available stroke of 0.035 inch/convolution was required. This also represented the bellows manufacturers recommended maximum stroke per convolution for flat plate bellows of this diameter. The alternate corrective action was to redesign the bellows. The bellows manufacturer submitted alternate designs to perform the function. However, all new designs also required approximately the same lengthening of the valve assembly as was required to accommodate the installed length of existing design revised to provide the 0.035 inch stroke/convolution. A new bellows weldment and poppet assembly were then fabricated with bellows heat treated to provide 0.035 inch stroke/convolution at the full length.

While the new bellows assemblies were being fabricated a test was performed to assure that the effective area of the bellows and the seat area were properly matched. The valve mechanism was assembled such that the installed position of the bellows left the poppet off the seat. Load increments were applied to the poppet and the poppet stroke measured, until the poppet seated. The resulting spring rate was calculated and agreed with prior test results (100 lb/in.). A load increment was added to the poppet and the inlet pressure was increased until the poppet lifted to relieve the pressure. This procedure was repeated with increasing load increments. The total seat load divided by the initial deflection of the bellows yielded a spring rate which was plotted against the pressure at which the poppet lifted to relieve the pressure. The characteristic correlated with all prior spring rate vs. pressure characteristics for this deflection and it was concluded that effective areas had been properly matched.

The new poppet assembly and bellows assembly were installed in the valve mechanism with new spacers to permit maximum convolution spacing. A force vs. stroke test at various system pressures was performed (Refer to Figures 16 and 17). The net spring rate was calculated at each pressure level and plotted against the system pressure. To facilitate comparison of test data with design analysis, the following value was obtained for each measured stroke:

$$\frac{F_{360} - F_0}{360 \text{ psi} \times A_{\text{eff nom}}} \times 100$$

Where

F_{360} = Force to produce specific deflection at 360 psi pressure

F_0 = Force to produce same deflection at 0 psi

A_{eff} = Nominal effective area of bellows

The resulting value indicates change in effective area over the operating pressure range if the bellows spring rate from 0 to 360 psig is constant, as was assumed in the design analysis (See Figure 18).

The net effective force, in addition to environment generated and preload forces, which the actuator must overcome to cause poppet motion, is therefore the combined values of spring rate and effective area change force, which in the original analysis was 9.5 pounds. Since it is impossible to isolate these two force characteristics by test, their combined effect is the controlling factor. In spite of the effective area change exceeding the assumed value used in design analysis, the total force required to achieve up to 0.077 inch of valve stroke at any pressure up to 360 psig did not exceed that, allowed for in the analysis (9.5 lbs).

Having determined the force characteristics of the assembly and established that with no poppet motion, the force required to maintain poppet position is stable within ± 0.5 lb, tests to determine leakage characteristics were performed. Prior to that time, some attempts were made to determine the leakage characteristics but all proved unsuccessful as the instability of poppet forces over the pressure range prevented accurate measurement of seat loads. From these attempts, however, it was evident that a series of break-in cycles, to "wear-in" the poppet and seal, improved the sealing characteristics.

The valve mechanism with the final bellows design (T13221/001A poppet assembly and T13222/003 bellows weldment) was installed in the test set up (See Figure 19). Initial leak tests were performed with GN_2 with leakage measured by water displacement. Seat loads up to 10 lb (4 lb/in.) were applied to the poppet and tests performed at pressures up to 100 psi. Excessive leakage was evidenced at all pressures greater than 5 psig. Leakage rate at any pressure was independent of applied seat load, indicating a fixed clearance existed across the poppet seat interface. The valve was disassembled and visually examined. The body seat evidenced radial surface scratches when viewed under 60X magnification. Seat molds were then made of the body seat while a seat lapping tool was fabricated. The seat molds were then examined on a 50X comparator as was the poppet seal profile. A composite of the profiles is shown in Figure 20. From this examination it was evident that inadequate Teflon protrusion prevented adequate Teflon squeeze to accomplish a good seal against a surface with radial scratches. The poppet seal was removed and a new Teflon seal installed. The finish cut on the Teflon was made to allow +0.0020, - 0.0015 inch more protrusion at the seal face. The body seat was lapped with 4-A grade lapping compound to remove all radial imperfections in the sealing area.

c. Complete Valve Assembly

The valve mechanism was then reassembled and the actuator mounted to the assembly to facilitate break-in cycling. The valve was mounted in the set up in an inverted position. In this orientation, dead weights could be suspended from a shaft attached to the armature to counteract the latching force thereby controlling and measuring seat load. Since the assembly of the actuator to the valve mechanism was accomplished without controlling the armature to pole face gap the latch force was measured and found to be 8.4 lb. The valve was then pressurized with GN_2 at 100 psig. Excessive leakage was measured and the valve was cycled from closed to open to closed ten times with no pressure applied. The 100 psig GN_2 was applied and though the leakage rate was reduced, it was still excessive. The valve was then cycled 50 times and GN_2 pressure again applied. There was no evidence of leakage at any pressure up to 360 psig for a period of 10 minutes. With 360 psig applied at the inlet and the outlet connected to a minimum ullage at 0 psig the valve was cycled from closed to open to closed and a leak check again performed. This procedure was repeated 10 times with no evidence of leakage at 360 psig for 3 minutes occurring after any cycle. The inlet pressure was reduced to zero and the valve left at ambient conditions for 15 hours. At the end of that period a leak check at 360 psig was performed with no evidence of GN_2 leakage for a period of 6 minutes.

The valve inlet was then attached to a regulated helium supply. With inlet pressure at 360 psig for 10 minutes, there was no evidence of leakage. The valve was then cycled 10 times as before, with a 3 minute helium leak check performed after each cycle. After the third and eighth cycle some leakage was observed and measured. The maximum rate measured over the three minute period was 18 scc/hr but the rate appeared to decrease with time. All other leak checks resulted in no measurable leakage. The valve was then disassembled and microscopically inspected.

A uniform and full circumference trace of the metal to metal contact was observed. The trace was characterized by a smoother appearing surface than surrounding areas but no marked deformation of the total seat area (depression of trace, distortion, or migration of material). There was no evidence of Teflon deposition on the seat resulting from sliding contact with the poppet seat. The trace was uniform width along its entire circumference indicating repeated, consistent contact with the poppet, hence good guidance. The poppet evidenced no significant change from its as-machined condition except significant amounts of foreign particulate contaminant in the relief gap between the upstream side of the Teflon seal and the retainer. The source of the contaminant was judged to be the helium regulator and supply line. The Teflon sealing surface evidenced no significant irregularities or damage as a result of exposure to the contaminants. No extrusion or change in contour of the Teflon was apparent.

The valve was then reassembled and the previous leakage test sequence performed. The valve again required a brief break-in series of cycles, but all leak checks performed evidenced zero or token leakage rates no greater than 10 scc/hr.

The valve was then installed in the water flow set up to verify performance under flow conditions (Refer to Figure 21). Poppet stroke was adjusted to 0.065 inch. The Δ ρ at rated flow vs. inlet pressure characteristic was then determined (See Figure 22). The valve was then actuated to the closed position with 18 Vdc while flowing 0.73 pps at a static inlet pressure of 360 psig. Zero liquid leakage was measured over a 20 minute period with the inlet maintained at 360 psig. The valve was again disassembled and inspected. Observations were essentially the same as the previous examination with the exception of a reduction in the quantity of foreign particulate contamination.

With the valve reassembled, helium leakage reduced to zero after a brief break-in cycle (25 cycles). Inlet pressure was then increased to 540 psig for 3 minutes. Leakage at this pressure was observed to be 1 bubble every 12 seconds. On reducing the pressure to 360 psig, leakage was zero over the 3 minute period at this pressure. This cycling of the inlet pressure was repeated 5 times with results essentially identical with prior observations.

The valve was then inverted in the set up and the seat load determined by suspending dead weights from a shaft extension. The inlet was then pressurized to 360 psig with helium and the leakage monitored. A dial indicator was employed to monitor poppet motion and one pound weight increments were applied to the shaft extension. After each weight increment addition, poppet motion was recorded and leakage was measured for three minutes. This was continued until the leakage rate exceeded 20 scc/hr. The actuator armature to pole face clearance was then adjusted to provide a different latch force and the loading procedure repeated. From the measured latch face, dead weight force at which leakage exceeded 20 scc/hr and the assumed seating diameter, the seat load rate (lb/in.) was calculated for correlation with the value used in the design analysis.

$$\frac{F_L - F_D}{\pi\, d_s} = \text{Seat load rate}$$

Where

F_L = Measured latch force

F_D = Dead weight load at which leakage just exceeds 20 scc/hr

d_s = Assumed seat diameter inches

Test A

$$\frac{8.4 - 3}{\pi \; Y \; 0.812} = 2.12 \text{ lb/in.}$$

Test B

$$\frac{13.2 - 5}{\pi \times 0.812} = 3.18 \text{ lb/in.}$$

With the valve assembly installed in the water flow set up the actuator was instrumented to determine valve response characteristics. Induced voltage signatures were recorded on an oscilloscope equipped with a Polaroid Camera at various input pressures and applied actuating voltages.

Over the pressure range of 0 to 360 psig and applied voltage range of 18 to 32 Vdc, opening response ranged from 0.006 to 0.018 seconds and closing response from 0.008 to 0.014 seconds (Refer to Figures 23 and 24). The performance of the magnetic reed switch position indicator was also verified again during these tests.

The final development test, prior to disassembly and inspection of the prototype, was to demonstrate the desirability of the ball joint poppet-shaft linkage. All prior testing had been performed with the ball free to rotate to accommodate misalignment of the shaft with the poppet. Since the prototype design did not control the alignment of the actuator with the valve assembly, it was logical to assume that the valve had performed satisfactorily without the need to align these two sections with any more accuracy than a visual examination. To demonstrate the desirability of the selected linkage, the bearing-ring was tightened against the ball to preclude free rotation. With the inlet pressurized and the valve closed, side loads were applied to the shaft. Under these conditions, it was impossible to achieve a closure that had a leakage rate within the specification allowable without a significant increase in poppet load. The selected design was therefore judged to be the most desirable configuration.

The valve assembly was then completely disassembled and inspected. The poppet-seat interface area showed no significant change in appearance beyond that noted after the first inspection of this area. The Teflon seal evidenced no significant change in profile or extrusion. The condition of the sealing surface showed evidence of burnishing with respect to the original machining marks, but no damage to the surface was observed in spite of the particulate contamination which the assembly had experienced during the test program. Sliding contact areas were examined to determine a potential galling problem. All showed evidence of metal to metal sliding contact, but evidence of galling, such as metal smears, burns or local burnished areas was nonexistent. Examination of other components evidenced no apparent wear or fatigue problems and development testing was concluded.

4. Summary of Design Changes Resulting from Development Testing

During the development phase of the program, the following design changes were incorporated in the prototype valve mechanism, and evaluated.

a. Teflon Sealing Profile

The initial prototype seal profile design required the machining of a double angle profile on the Teflon seal. Further analysis and experience with this profile gained in the manufacture of Apollo engine injector valves, revealed a potential stable seating diameter problem due to cold flow of the Teflon. A single angle profile, with a 5° differential angle with the seat surface, was selected. Initial development testing to verify leakage rate characteristics resulted in the requirement to increase Teflon protrusion + 0.0020, - 0.0015 inch to assure adequate squeeze to accomplish the seal.

b. Actuator Magnetic Path

As presented in Section 3.0 a, an analytical error in the design of magnetic path areas was disclosed by a more rigorous review of the design analysis. The corrective action implemented during development testing was consistent with the corrected analysis.

c. Bellows Installed Length

The initial development test to determine force-stroke characteristics of the bellows at pressures, revealed a significant interrelationship between spring rate, pressure and convolution spacing. The bellows manufacturers contacted at the start of the program offered no information nor claimed any knowledge of this imposing any restriction. As a result of the analysis of initial test results, a minimum installed length to assure stable characteristics was determined and verified by future test. This new length restriction required an increase in valve length and consequently a unit weight increase.

5. Design Support Evaluation Tests

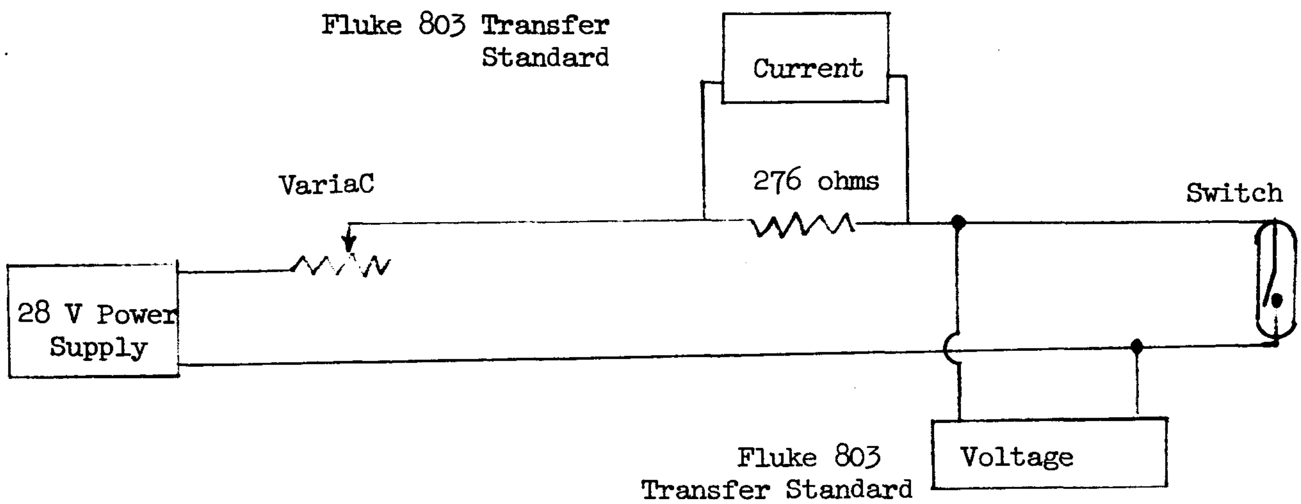
In support of the initial design concept, three evaluation test programs were initiated to verify specific configurations. Questionable design areas were evaluated, and test results are:

a. Contact Resistance of the Selected Magnetic Reed Switch Position Indicator

Data available from suppliers of magnetic reed switches indicated a marginal ability to meet the 50 milliohm contact resistance requirement and a reluctance to guarantee this characteristic due to wide variations in test methods. Two of the SPDT switches procured for the development test were tested to determine contact resistance with up to 2.5 watts applied across the switch leads at constant current. The two switches were then connected in parallel and the test repeated. Results were as follows:

	N. C. Contacts	N. O. Contacts
Switch No. 1	38 to 49 milliohms	33 to 39 milliohms
Switch No. 2	30 to 36 milliohms	30 to 36 milliohms
Switches 1 and 2 in parallel	11 milliohms	11 milliohms

The test set up is shown schematically below:



The tests verified contact resistance to be well within the 50 milliohm maximum limit.

b. <u>Aluminum Coil Wire Junction to Dissimilar Leads</u>

Two design approaches were evaluated. The ability to join the aluminum leads to the 52 alloy header pins was evaluated and the joining of copper leads to the aluminum at the coil was evaluated. Samples of aluminum lead wire and the 52 alloy header pins were plated with copper, silver and nickel and those with similar plating were joined with 50% zinc solder using standard soldering techniques. The silver and nickel plated aluminum wires were extremely brittle and eventually fractured at the termination of the plating. These joints also exhibited joint resistance in excess of 50 milliohms. The copper plated samples yielded a joint of less than 1 milliohm resistance but embrittlement posed a problem unless complete restraint of the lead could be assured by the design.

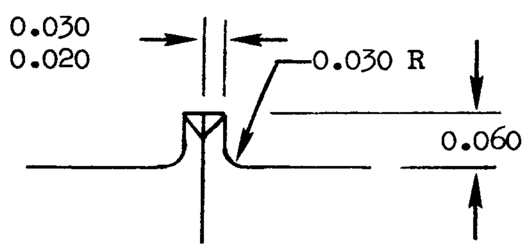
Additional wire and pin samples were joined using the "Metahese" process, a proprietary plating technique of Camwil, Inc., of Los Angeles. The samples exhibited contact resistance similar to the soldered joints and embrittlement of the plated areas resulted in ultimate fatigue failure of the wire.

The second design alternate was evaluated by obtaining a low resistance copper to aluminum wire joint using 90% tin, 10% zinc solder, applied with an ultrasonic soldering pot. By providing one turn of the copper lead before bringing the pigtail from the coil, the structural integrity of the joint was assured by the final coil wrap of mylar tape.

The second design alternate was adopted on the basis of its simplicity.

c. Weld Joint and Configuration

Fusion welds were selected for the deliverable designs. To assure that the "Marq metal" actuator housing could be thus welded to the 321 cres valve body, samples were fabricated featuring the proposed burn-down lip. Thinner section lip samples were also fabricated. Samples were welded such that two dissimilar metal fusion joints and a "Marq-metal" to "Marq-metal" fusion joint was accomplished. The samples were then examined for penetration, heat affected zone and weld quality. Based upon this examination the configuration shown below was selected.



Configuration Shown Before Fusion Weld

All joints (except tube connection and bellows welds) exposed to pressures are threaded before closure fusion welding to assure structural integrity of the joint.

C. Deliverable Design

The incorporation of all demonstrated prototype features into a minimum weight unit meeting envelope and performance requirements of the work statement, was the primary objective of the deliverable design effort. The design changes resulting from development testing and the configurations selected on the basis of the sample evaluation tests were incorporated. The resulting design is presented in Figure 25.

In addition to selecting wall thicknesses to perform specific functions, the wall thickness and joint design of unwetted portions of the valve were selected to withstand valve fluid pressures, to guarantee containment of fluids in the event of a bellows failure. Magnetic materials not inherently compatible with the operating fluids are heavily electrolized to provide wear resistance, and resistance to attack.

Surface finishes were specified commensurate with required function. Sliding wear surfaces and the seating surfaces are finish ground to an 8 rms finish before electrolize is applied. Critical flow path surfaces are contoured with generous transition radii, wherever possible, to minimize turbulence and surface finishes are at least 125 rms.

Except for final closure welding, the sequence of assembly and techniques employed are identical with those established during build up and development of the prototype unit. The weld joint design was verified by sample testing.

A summary of estimated and actual weight of components and sub-assemblies of the valve is shown in Table III, below.

TABLE III

VALVE WEIGHT BREAKDOWN

P/N	Name	Preliminary Estimated Weight (lbs)	Actual Weight (lbs)
X22098	Lower Body Assembly	0.2630	0.233
X22726	Bearing Ring	0.0009	0.001
X22725	Bellows Assembly	0.0465	0.067
X22728	End Plate	0.0048	0.005
X22727	Lock Screw	0.0075	0.005
X22718	Upper Body	0.1080	0.087
X22719	Poppet Assembly	0.0664	0.066
X22480	Actuator Housing	0.4970	0.396
X22701	Coil Assembly - Closing	0.0166	0.066
X22706	Magnet	0.0496	0.049
X22483	Armature	0.0880	0.092
X22703	Coil Assembly - Opening	0.0408	0.140
X22709	Potting Shell	0.0078	0.008
X22707	Cover Assembly	0.0405	0.048
X22482	Pole Piece	0.0970	0.131
X22713	Shaft	0.0108	0.011
X22712	Switch Mount Bracket	0.0010	0.001
X22705	Shim	0.0106	0.009
Misc. Standard Parts & Pigtail		0.1600	0.162
X22700	Valve Assembly	1.512	1.565

The weight of the first deliverable design, though essentially within the 1.50 lb limit of the specification does not represent the minimum attainable weight of the assembly. With adequate time to perform a more rigorous stress analysis and at a greater unit cost, weight savings could be achieved by further contouring of the valve body. Also, the nominal valve body wall thickness of 0.070 inch can be reduced in certain areas, and the contouring designed to achieve a more consistent wall thickness. The joining of the two body halves could be accomplished by electron beam welding, thus reducing the weight due to the threaded joint and transition into the weld lip. Additional weight savings may also be achieved by electron beam welding of the actuator housing to the pole piece and cover assembly to pole piece. The use of mounting lugs rather than a welded bracket also would result in a weight savings.

Utilizing a lighter weight pigtail, it is estimated that an ultimate unit weight of 1.30 lbs can be achieved. The delivered pigtail is a shielded jacket cable selected due to prototype development handling requirements. It weighs 0.137 lbs. An unshielded pigtail suitable for flight usage is estimated to weigh only 0.07 lbs.

Excluding the weight of the pigtail, it is estimated that an ultimate unit weight of 1.21 lbs can be achieved.

After completion of all welding operations on the valve section, proof pressure and seat leakage tests were performed. These tests all indicated satisfactory sealing characteristics. The actuator sections were then built up, welded in place, and final electrical connections made. A seat leakage check following completion of the assembly, yielded leakage rates in excess of the allowable. Removal of the cover assembly to provide access to the shaft-armature interface permitted resetting the closed position armature air gap to achieve a proper poppet seal. The air gap shift resulted from the thermal cycling during welding and potting cure operations. During these operations the valve had been in the closed position and "cold flow" of the Teflon seal during the 300°F potting cure bake cycle was suspected. Therefore, to aid the recovery of the proper Teflon profile and also test the stability of the armature air gap, the units were baked at 250°F for four hours with the valve in the open position, and all subsequent assembly operations performed with the valve in the open position.

It is recommended that a valve be maintained in the open position whenever it is not functioning. Valves fabricated under this contract have experienced up to 5000 cycles of operation without a seal failure, but additional testing will be required to demonstrate a life of at least 20,000 cycles.

Final acceptance testing yielded pressure drops in excess of specification allowables (2.55 to 2.90 psi vs. 2.0 maximum allowable). A rigorous study of the prototype flow performance curves and flow passage differences between the prototype and the deliverable units was undertaken. The pressure drop and stroke of the deliverable units did not correlate with any point on the curves generated during the prototype testing. Some differences however, do exist between the prototype and deliverable flow paths. In adhering to the envelope restriction of 0.88 inch maximum from tube centerline to valve base, the clearance between the inlet bellows flange (lower) and the poppet guide spyder is less than that found to be optimum during prototype testing. Also, the depth of the milled outlet channel is greater than that of the prototype, resulting in a more tortuous flow path. A reduction in pressure drop, to within specification allowable, is possible on future units by reducing the length of straight full diameter tubing at the inlet and outlet to permit a more gradual transition from the body interface to the tube-centerline envelope restriction. This would permit a reduction in the depth of the outlet channel. A change in the technique of installing the inlet bellows which would reduce unit weight would also permit increasing the flange to spyder clearance with a subsequent reduction in pressure drop.

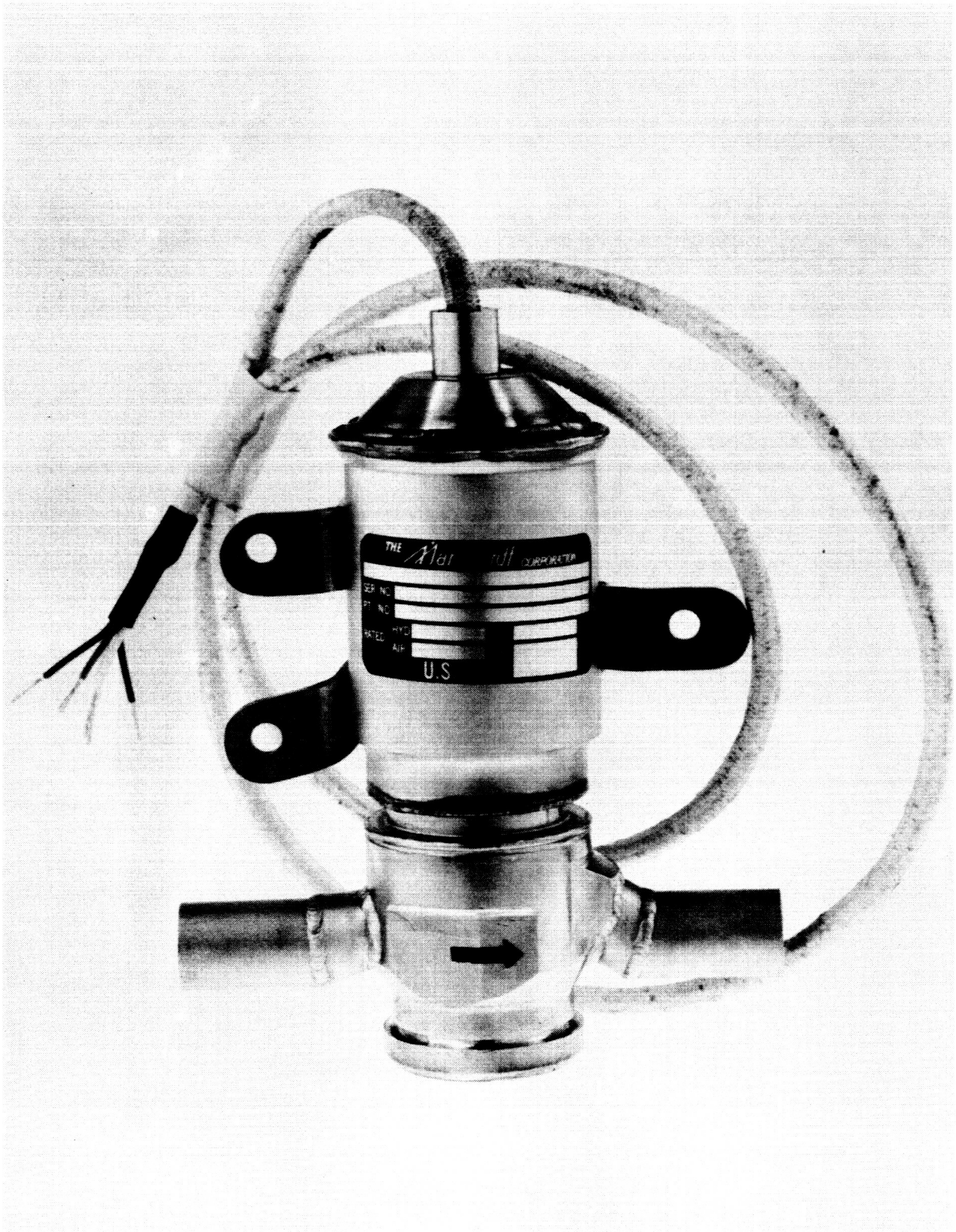
The performance of the position indicator switches is also questionable. The reed switches failed to respond to armature position as had been the case during prototype testing. A microswitch manufactured by the Texas Instrument Co., was investigated, but the long lead time precluded their use on the deliverable units. To provide valve position monitoring, a single reed switch is used and a permanent magnet is employed to bias the switch such that armature position will properly actuate the switch. Since time did not permit a thorough testing of this installation, little confidence is placed in this feature. Future units will employ the microswitch mentioned above.

The installation of this microswitch can be accomplished without modifying the envelope or magnetic path areas, provided a suitable actuating device can be employed to prevent excessive switch button overtravel.

Table IV presents a summary of the significant performance criteria, the value demonstrated by the first deliverable units, and proposed improvements, where required, to meet the specification requirements on future valves.

TABLE IV

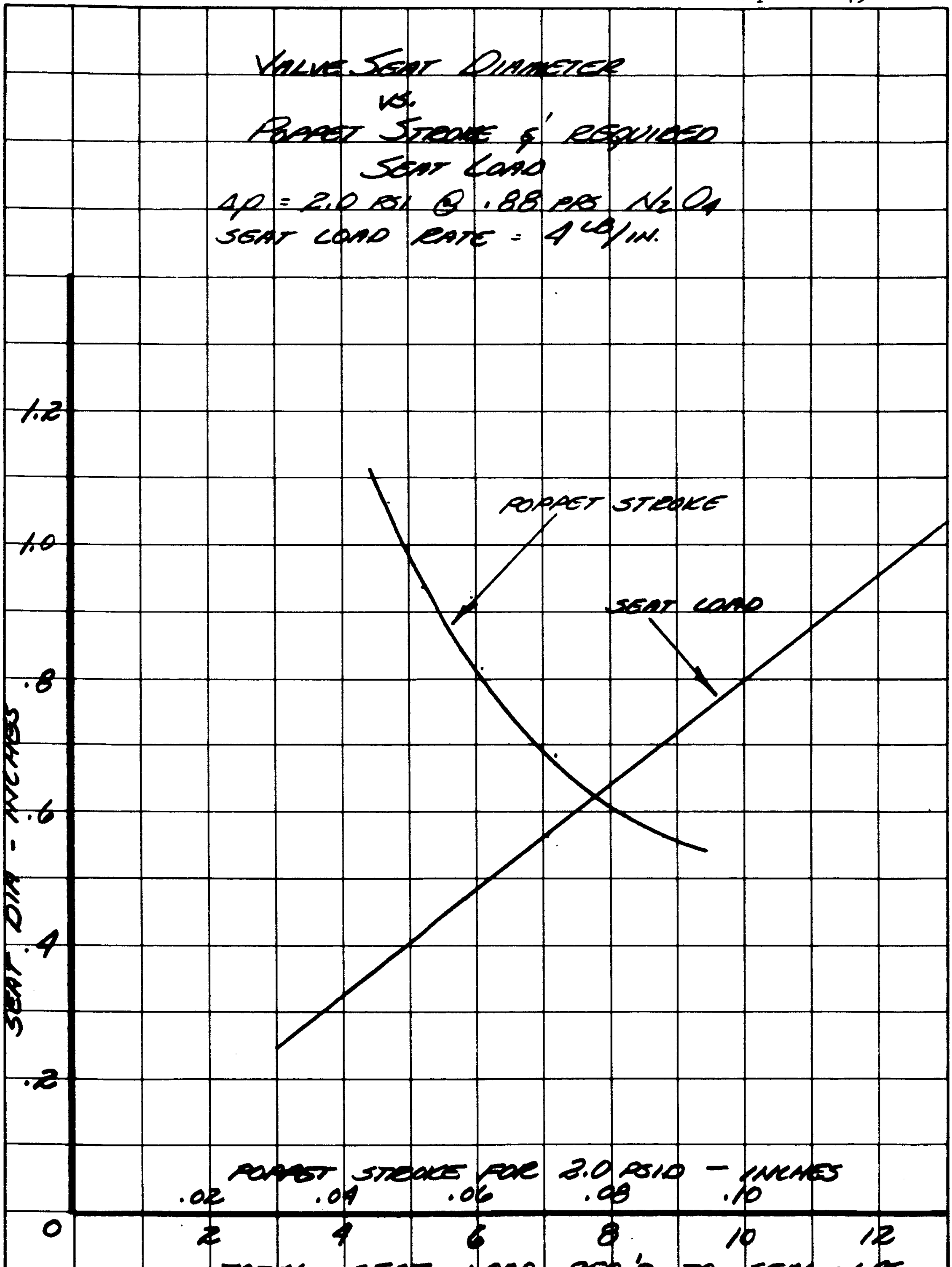
Specification Requirement	First Deliverable Lot Demonstrated Value	Proposed Improvements
1. 2.0 psi max ΔP at flow rate of 0.88 pps N_2O_4	2.55 to 2.90 psi	Reduce depth and increase width of outlet channel; improve area transitions
2. 1.5 lbs max unit weight	1.565 lbs	Contour body; butt weld mounting lugs; light weight pigtail
3. Position indicator to monitor valve position	Marginal-low confidence technique used	Use miniature microswitch
4. 20 scc/hr max helium internal leakage at 0 to 360 psi	0 scc/hr	
5. Minimum pull in voltage not to exceed 18 vdc	14 vdc at 360 psi 12 vdc at 0 psi	



X22700 VALVE ASSEMBLY - BISTABLE - SHUTOFF (U)
28 DEC 65

NEG. 7014-1

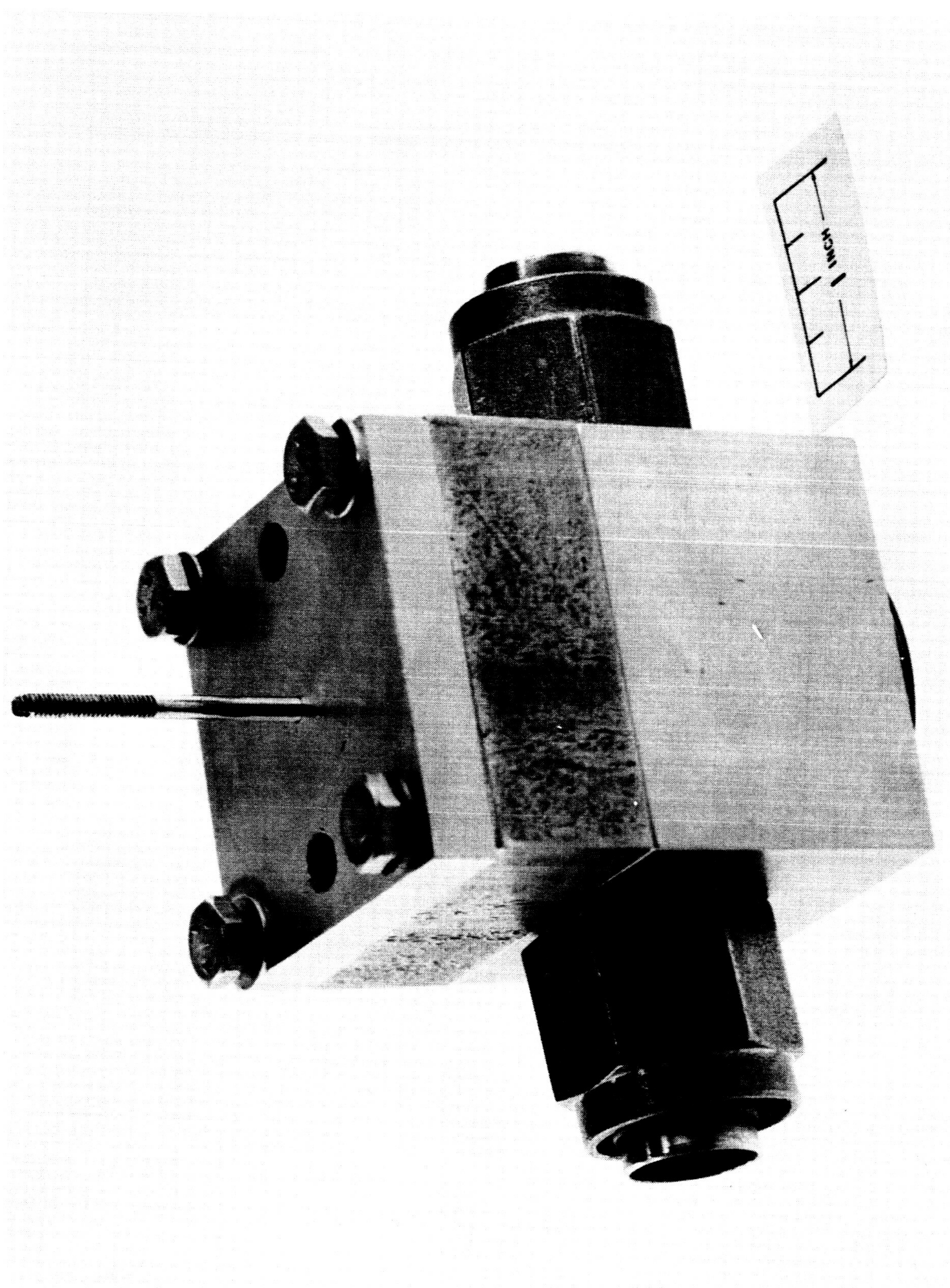
FIGURE 1



TMC A1668

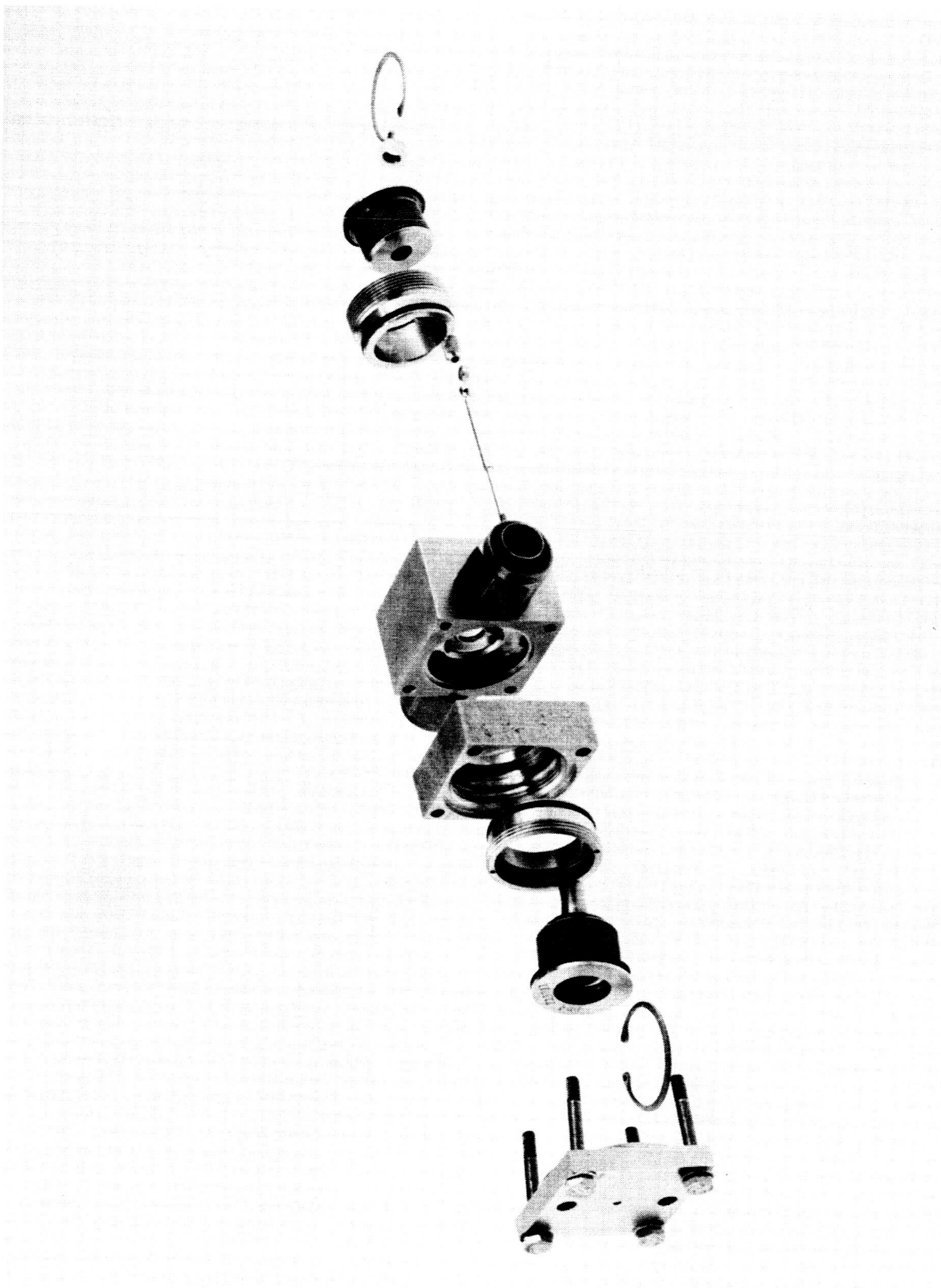
FIGURE 2

NEG. 6817-3



T-13228 PROTOTYPE VALVE MECHANISM (U)
5 AUG 65

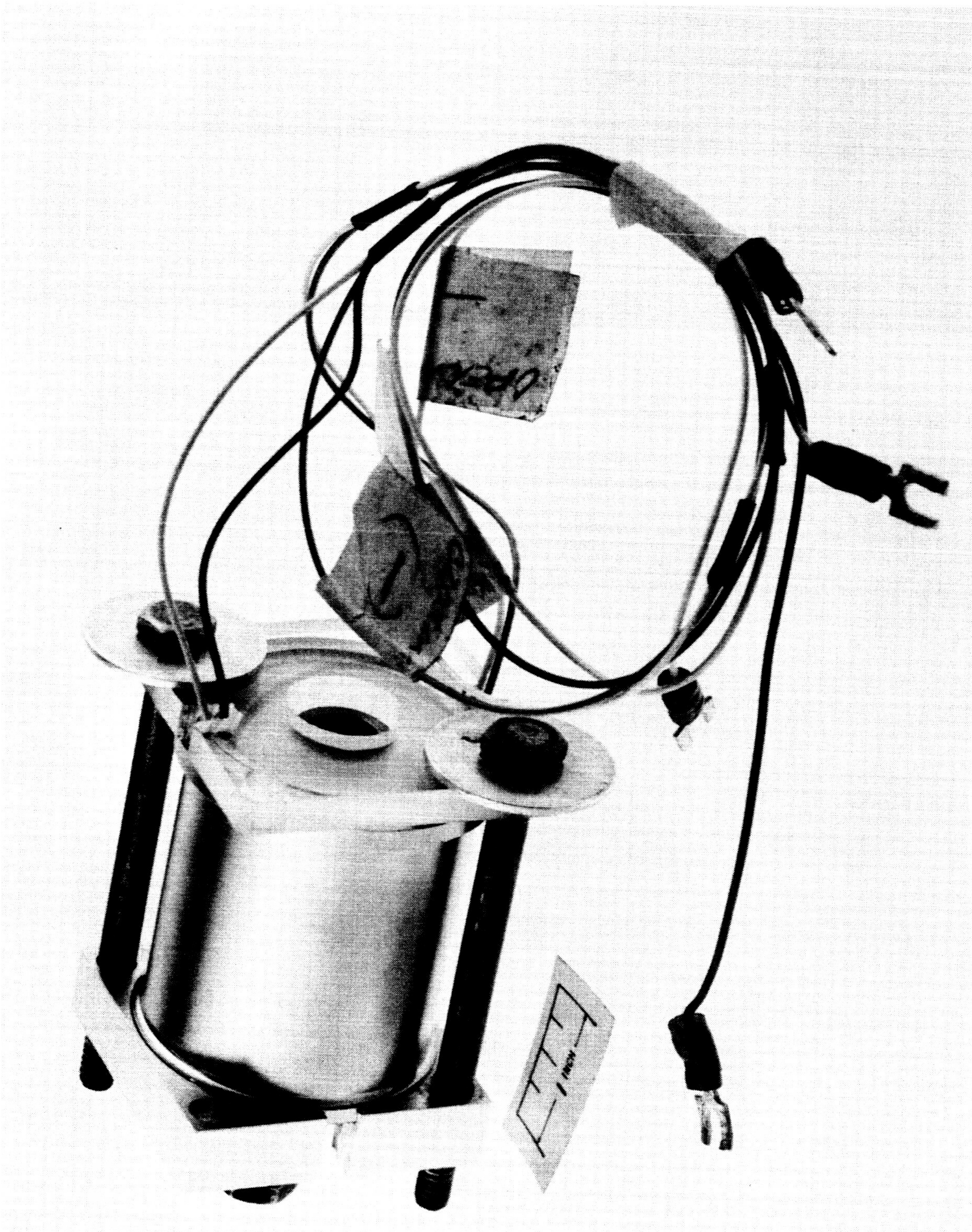
FIGURE 3



NEG. 6817-1

EXPLODED VIEW - T-13228 PROTOTYPE VALVE MECHANISM
5 AUG 65 (U)

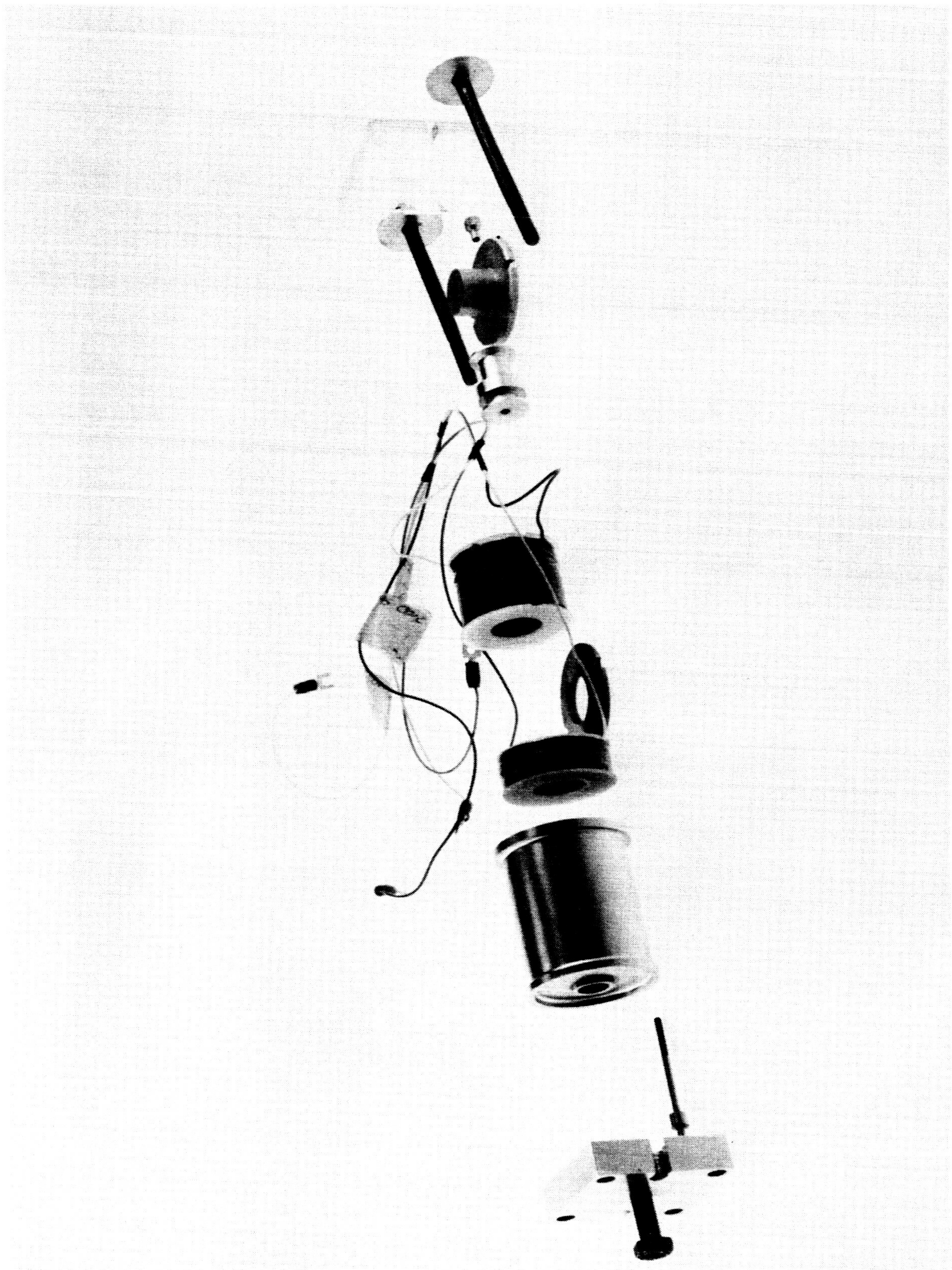
FIGURE 4



NEG. 6817-4

T-13228 PROTOTYPE ACTUATOR ASSEMBLY (U)
5 AUG 65

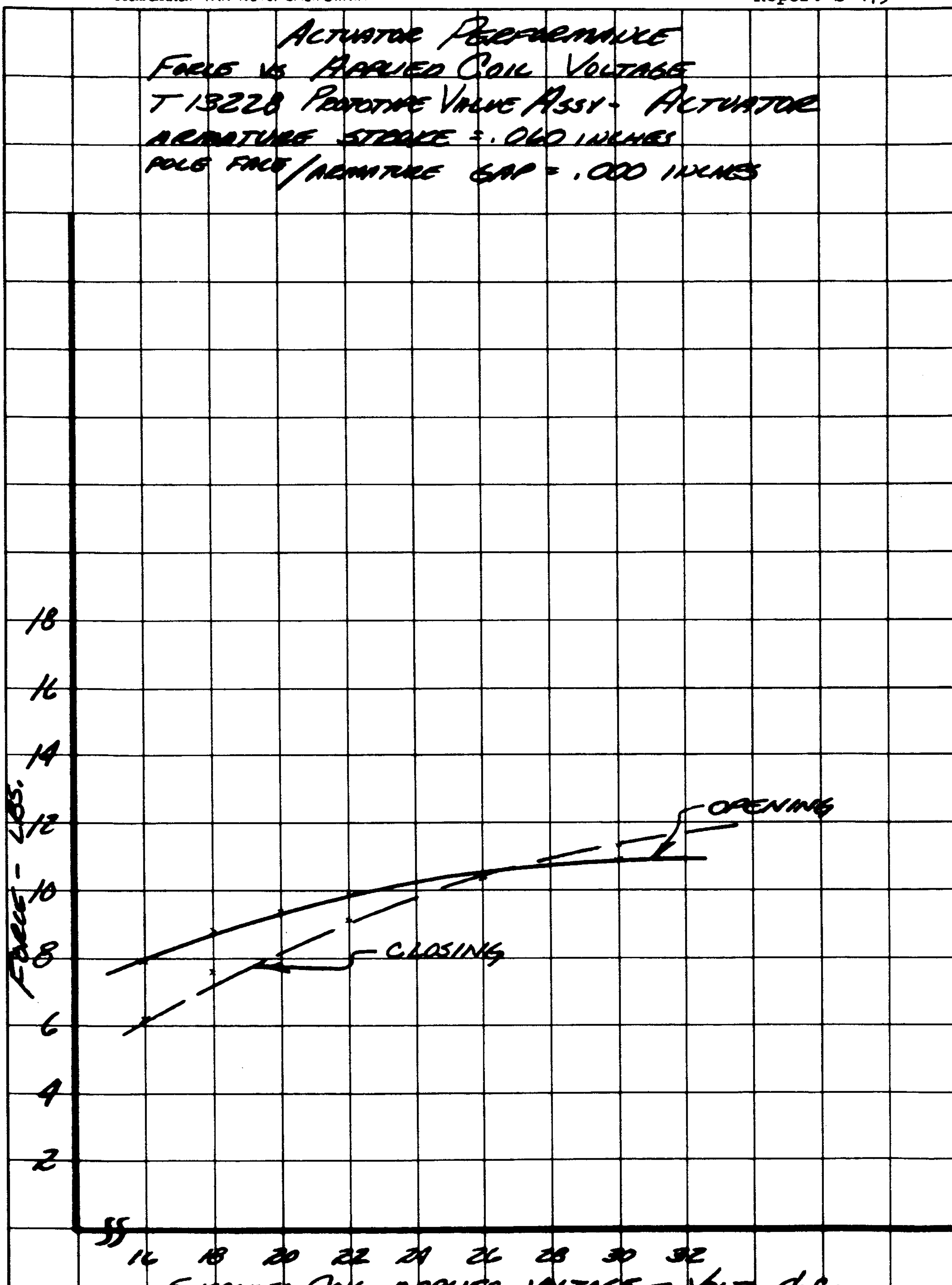
FIGURE 5



NEG. 6817-2

EXPLODED VIEW - T-13228 PROTOTYPE ACTUATOR ASSEMBLY (U)
5 AUG 65

FIGURE 6



TMC A1668

FIGURE 7

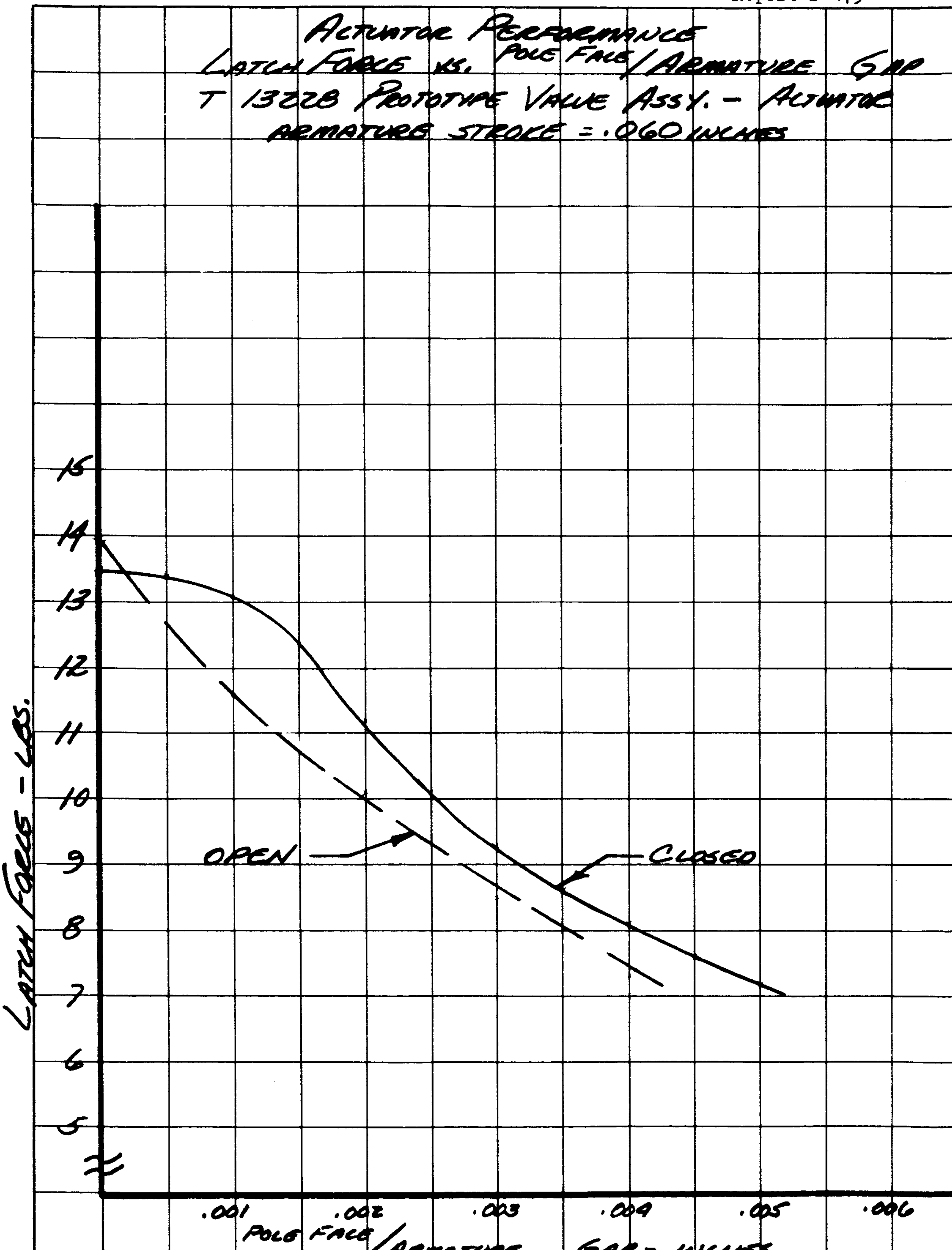
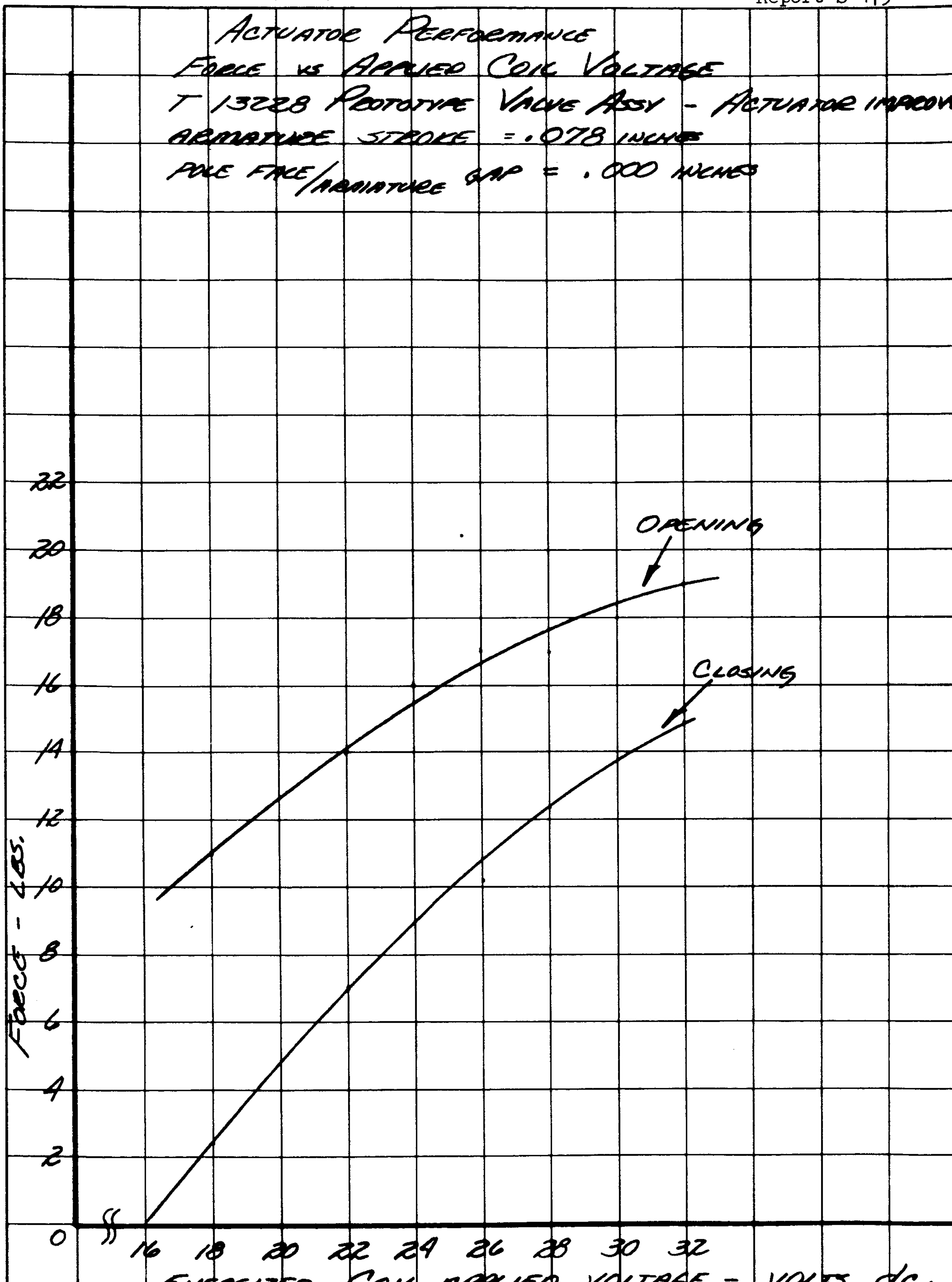

ACTUATOR PERFORMANCE
LATCH FORCE VS. POLE FACE/ARMATURE GAP
T 13228 PROTOTYPE VALVE ASSY. - ACTUATOR
ARMATURE STROKE = .060 INCHES
LATCH FORCE - LBS.
15
14
13
12
11
10
9
8
7
6
5
OPEN
CLOSED
.001
.002
.003
.004
.005
.006
POLE FACE/ARMATURE GAP - INCHES

FIGURE 8

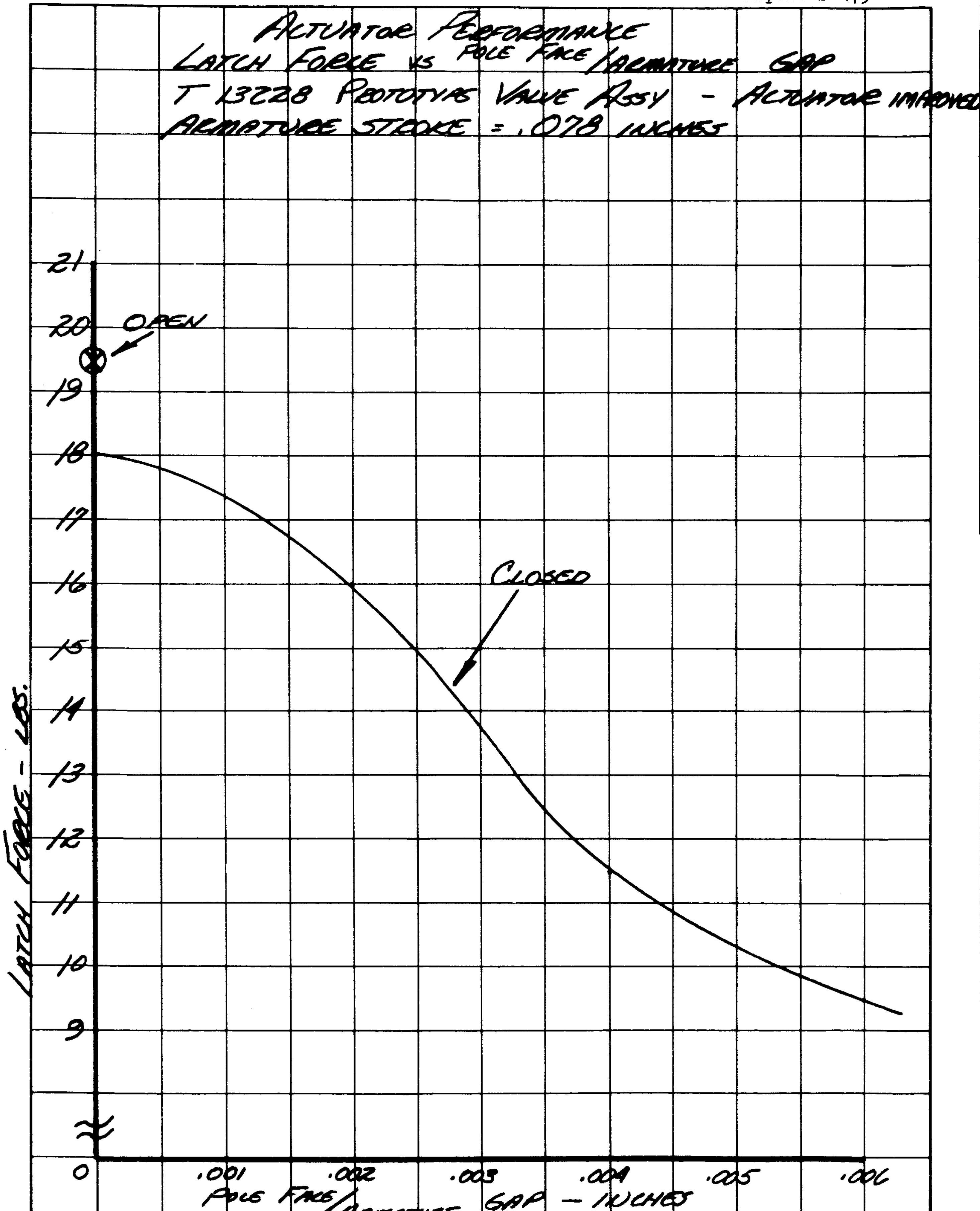

ACTUATOR PERFORMANCE
FORCE vs APPLIED COIL VOLTAGE
T 13228 PROTOTYPE VALVE ASSY - ACTUATOR IMPROVED
ARMATURE STROKE = .078 INCHES
POLE FACE/ARMATURE GAP = .000 INCHES
OPENING
CLOSING
FORCE - LBS.
0
2
4
6
8
10
12
14
16
18
20
22
16
18
20
22
24
26
28
30
32
ENERGIZED COIL APPLIED VOLTAGE - VOLTS dc.

THE Marquardt CORPORATION VAN NUYS, CALIFORNIA

PREPARED BY R. LYNCH

DATE 5-4-65

Report S-479



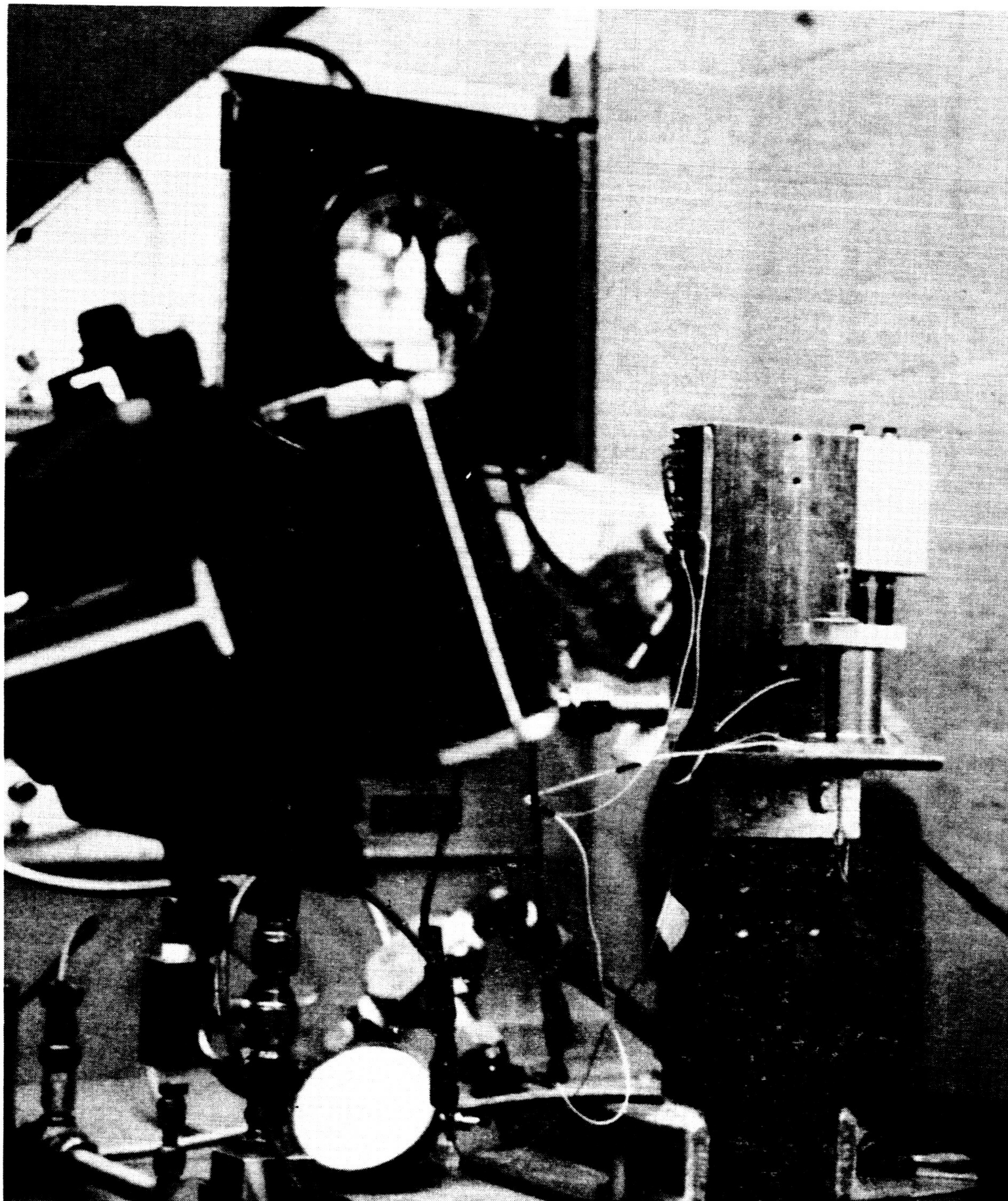
TMC A1668

FIGURE 10

SETUP - ACTUATOR TEST
(COPY)20 DEC 65 (U)

NEG. C7001-1

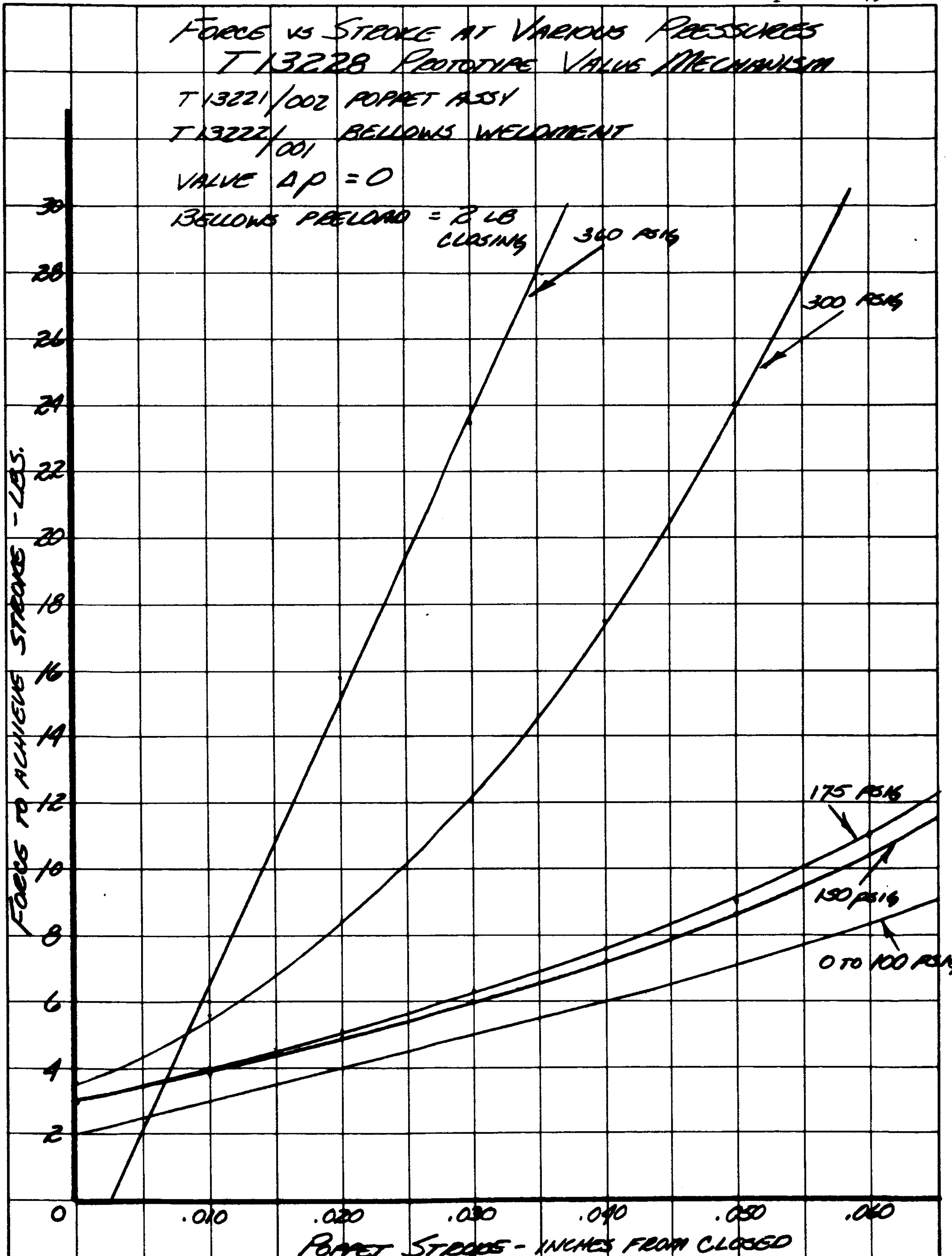
FIGURE 11



NEG. C7001-2

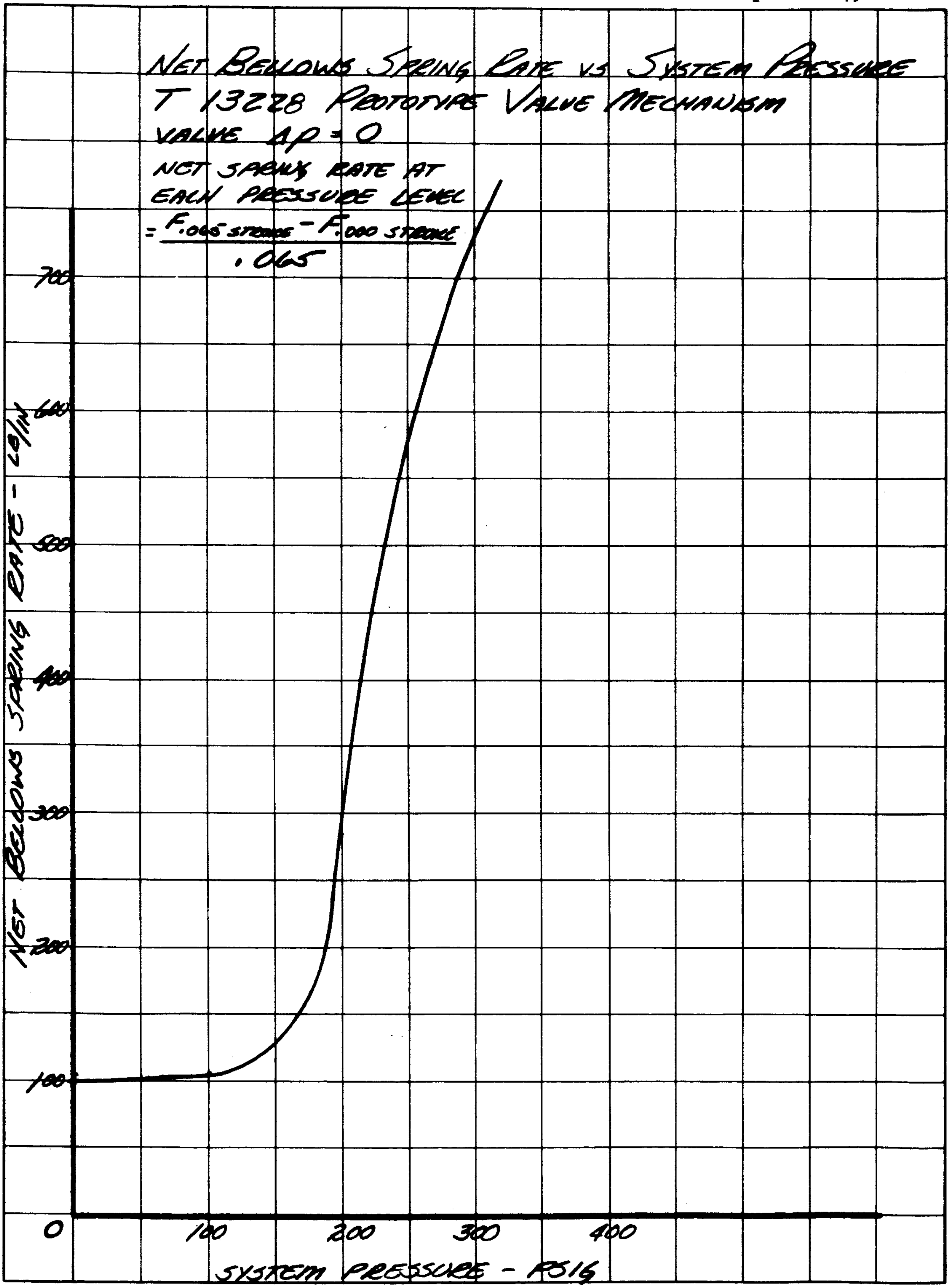
SETUP - ACTUATOR TEST
(COPY)20 DEC 65 (U)

FIGURE 12



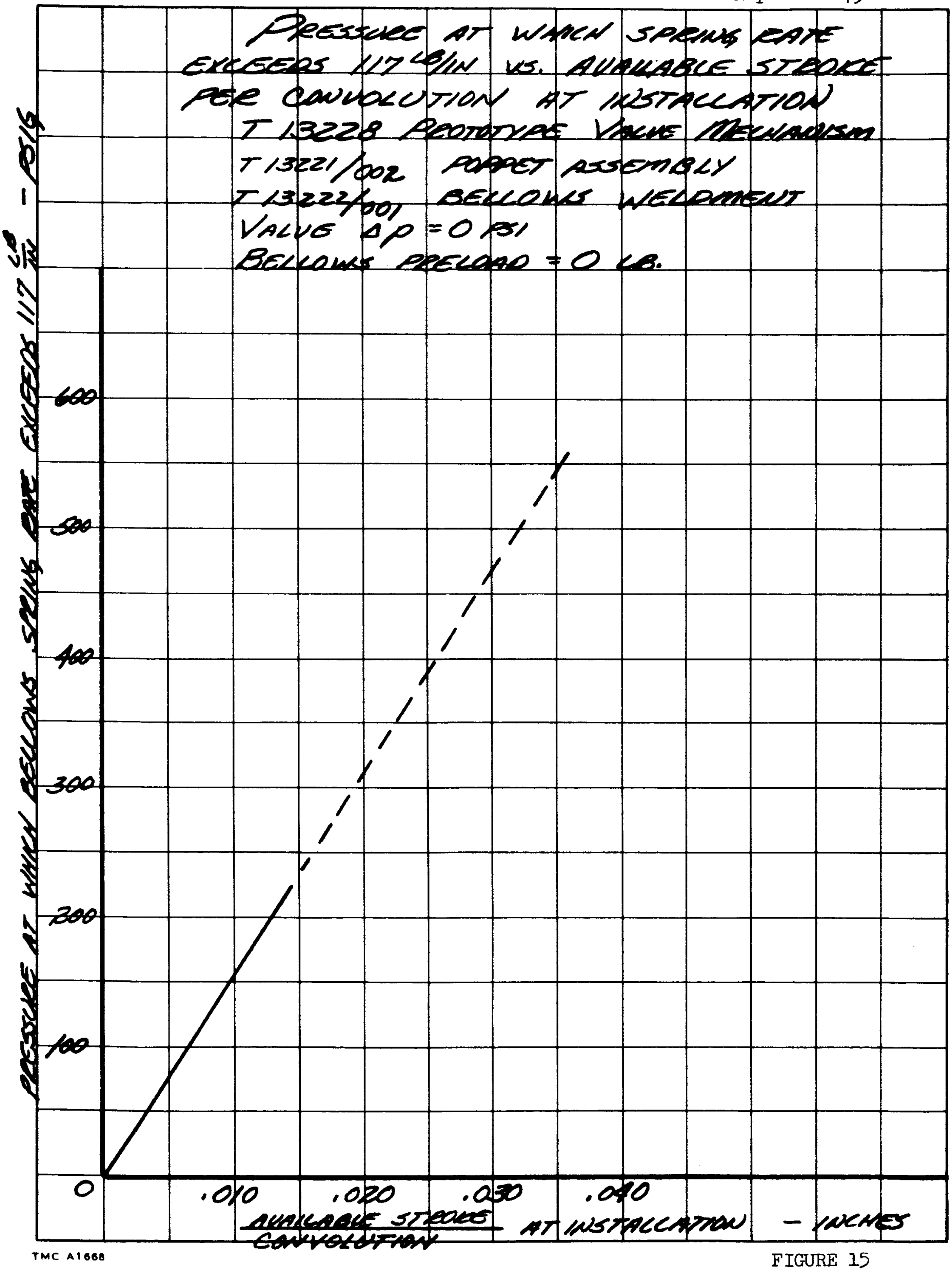
TMC A1668

FIGURE 13



TMC A1668

FIGURE 14

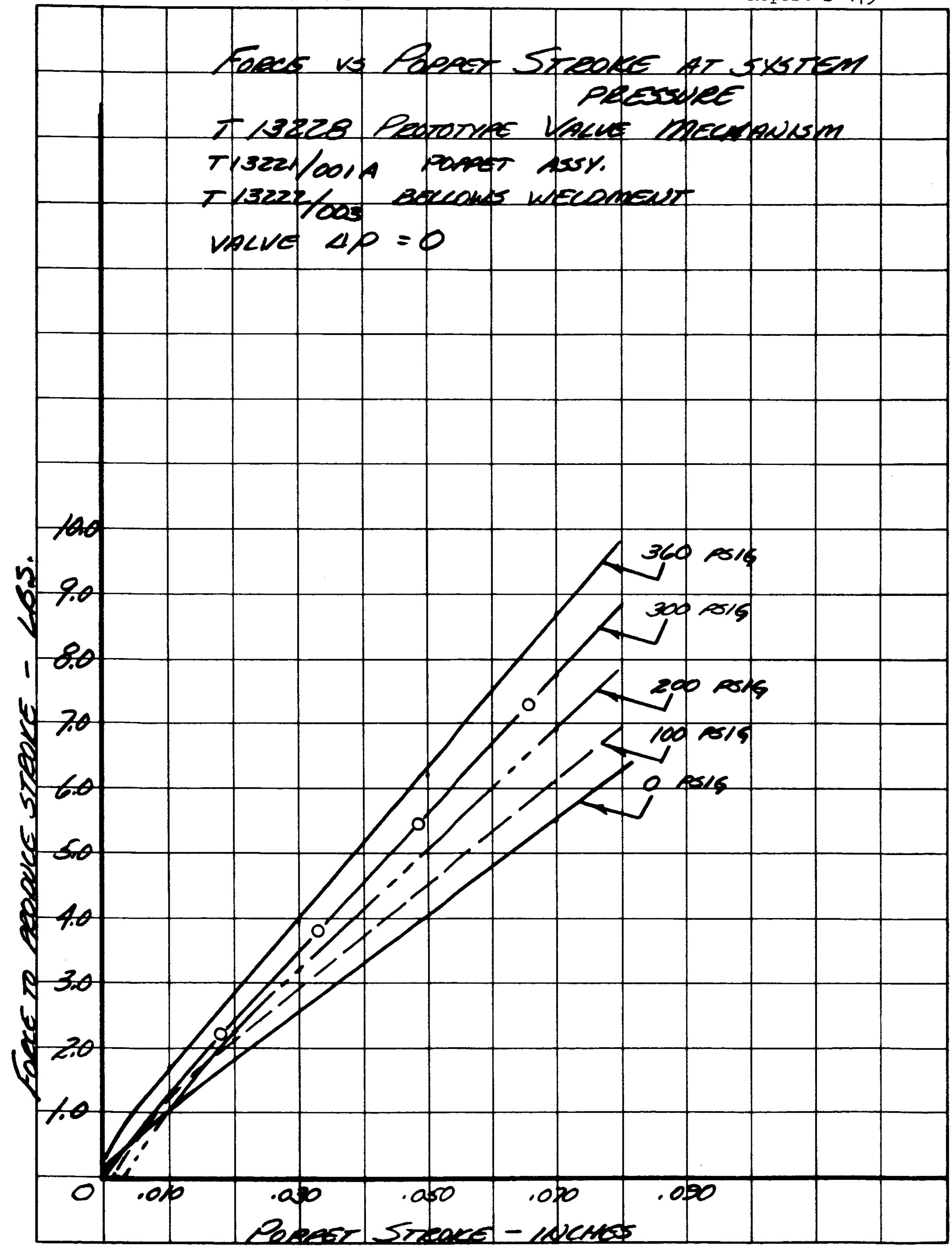


TMC A1668

FIGURE 15

THE Marquardt CORPORATION VAN NUYS, CALIFORNIA

PREPARED BY R. LYNCH
DATE 3-2-65
Report S-479



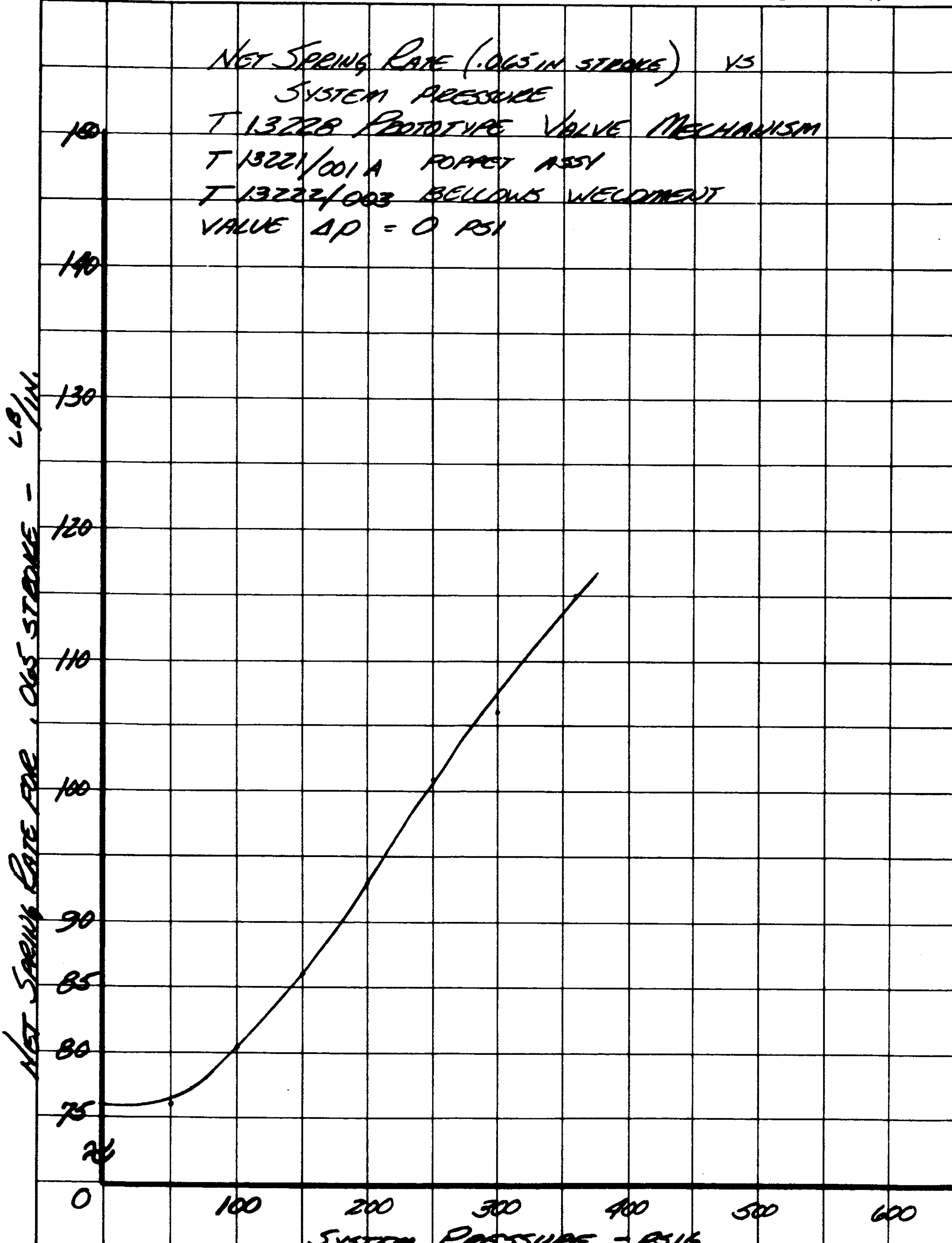
TMC A1668

FIGURE 16

THE Marquardt CORPORATION VAN NUYS, CALIFORNIA

PREPARED BY R. LYNCH
DATE 8-3-65

Report S-479



TMC A1668

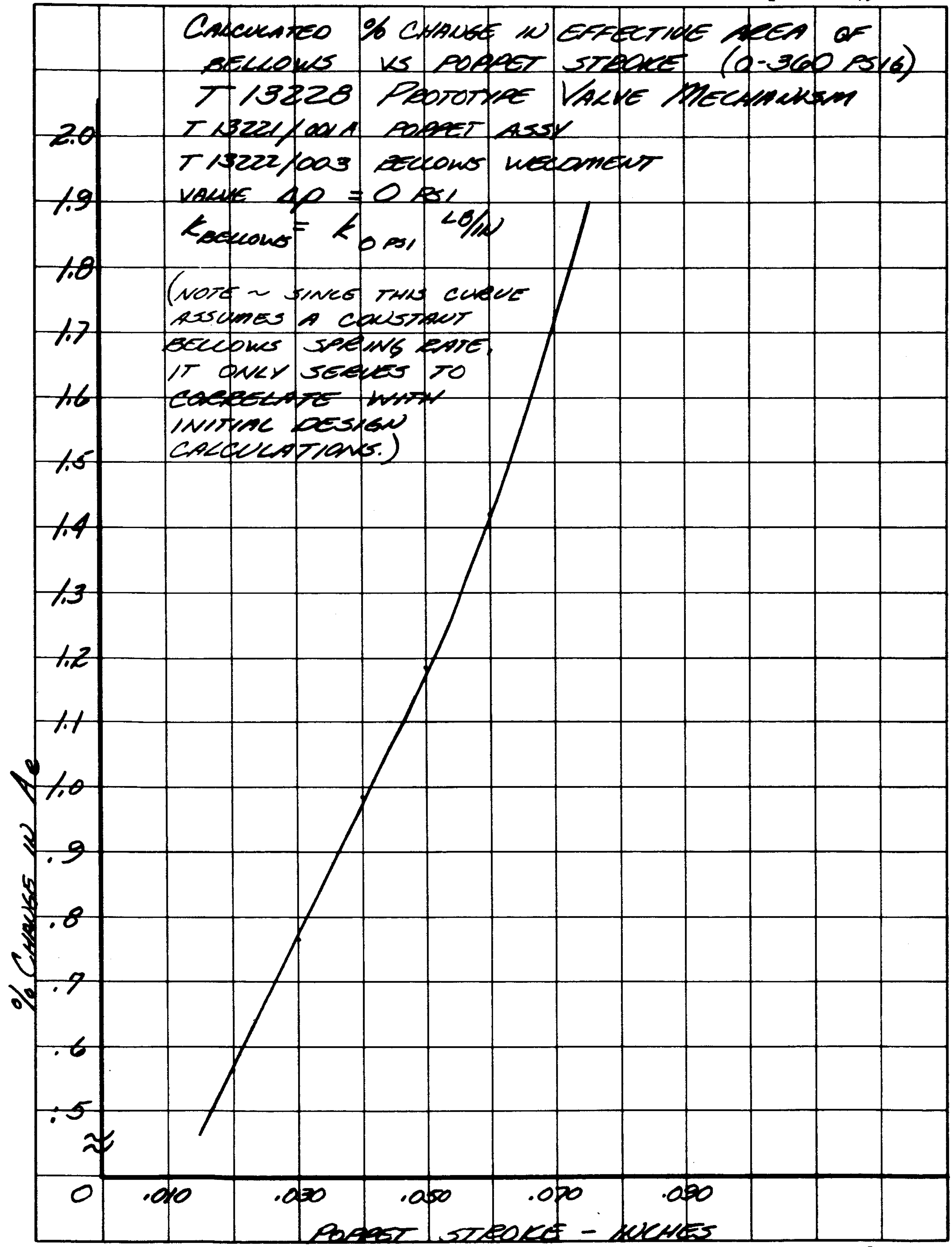
FIGURE 17

THE Marquardt CORPORATION VAN NUYS, CALIFORNIA

PREPARED BY E. LYNCH

DATE 8-10-65

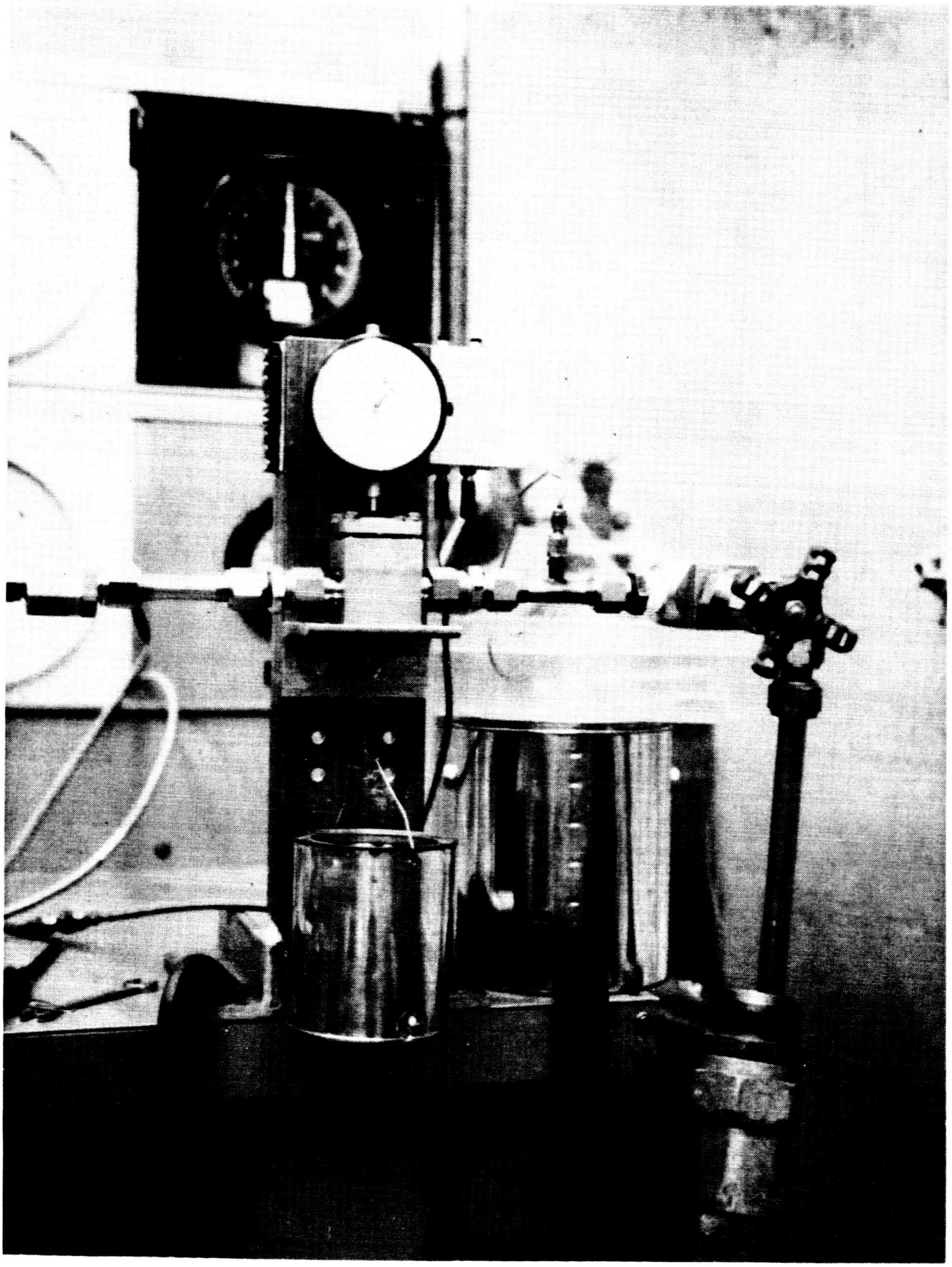
Report S-479



TMC A1668

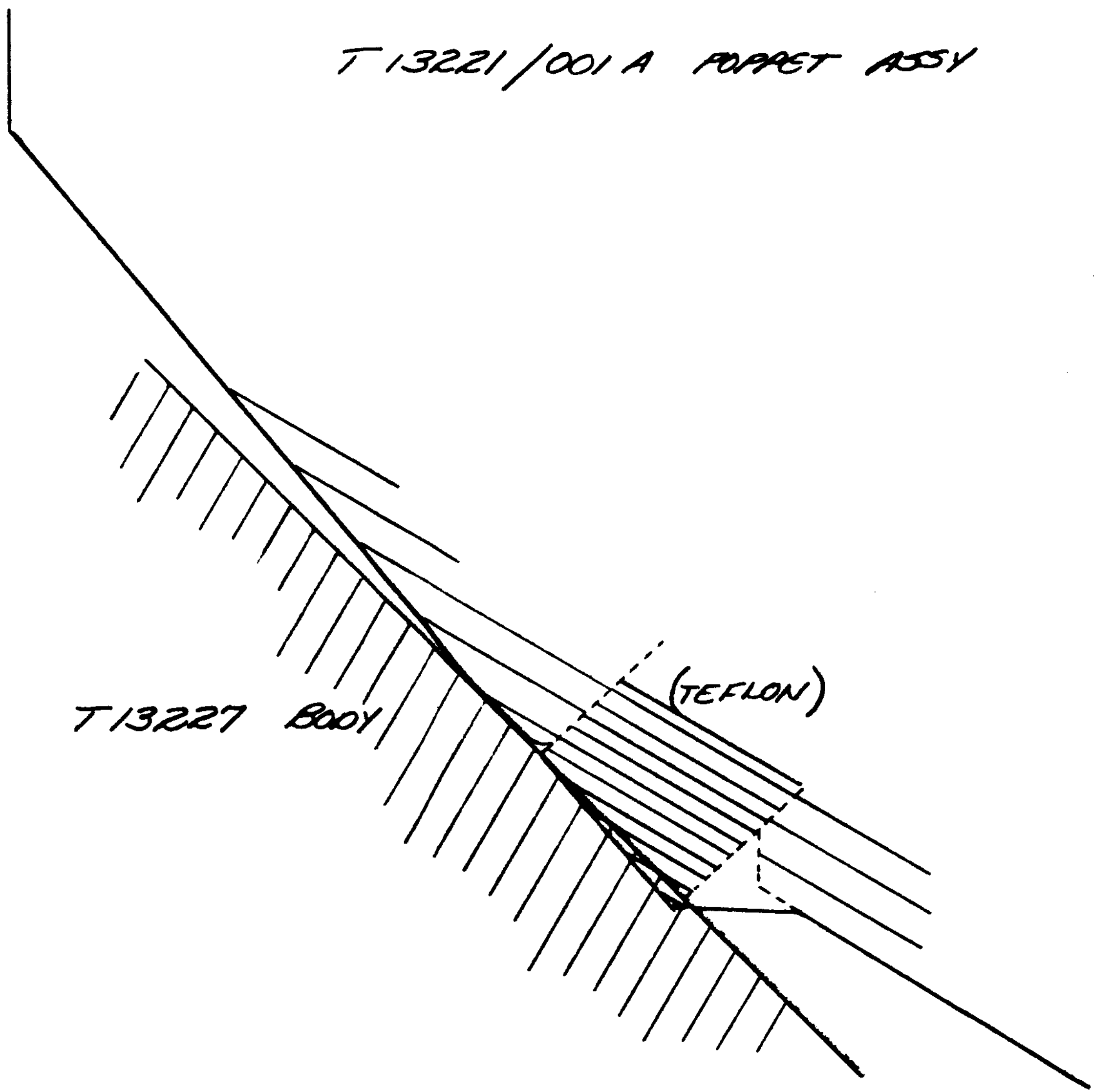
FIGURE 18

NEG. C7001-4



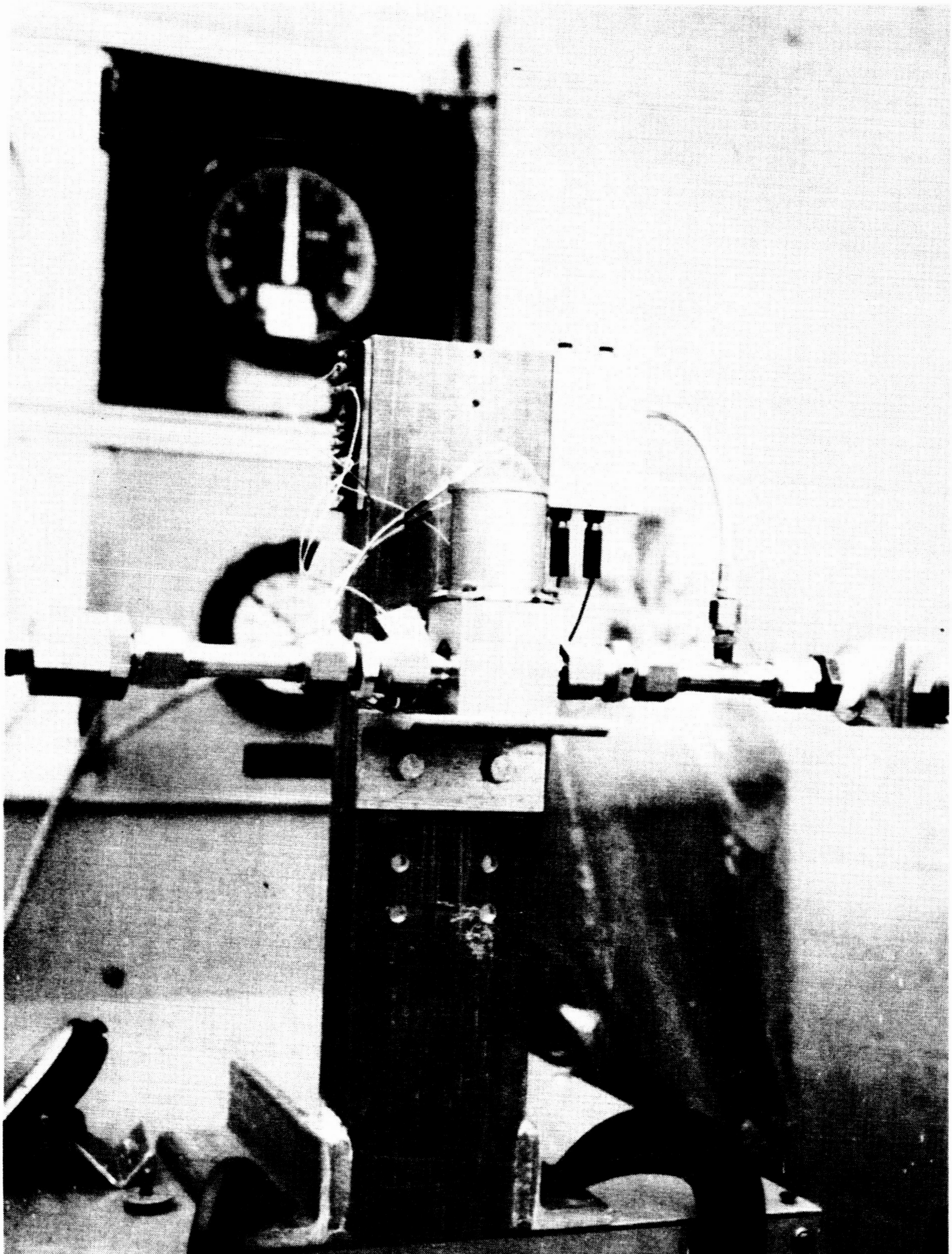
SETUP - VALVE SECTION FORCE vs. STROKE TEST
(COPY)20 DEC 65 (U)

FIGURE 19



COMPOSITE 50X COMPARITOR PROFILE
T 13227 BODY SEAT MOLD
T 13221/001 A POPPET ASSY (SEAL AREA)
DOTTED DETAIL & CROSS HATCHING ADDED FOR CLARITY

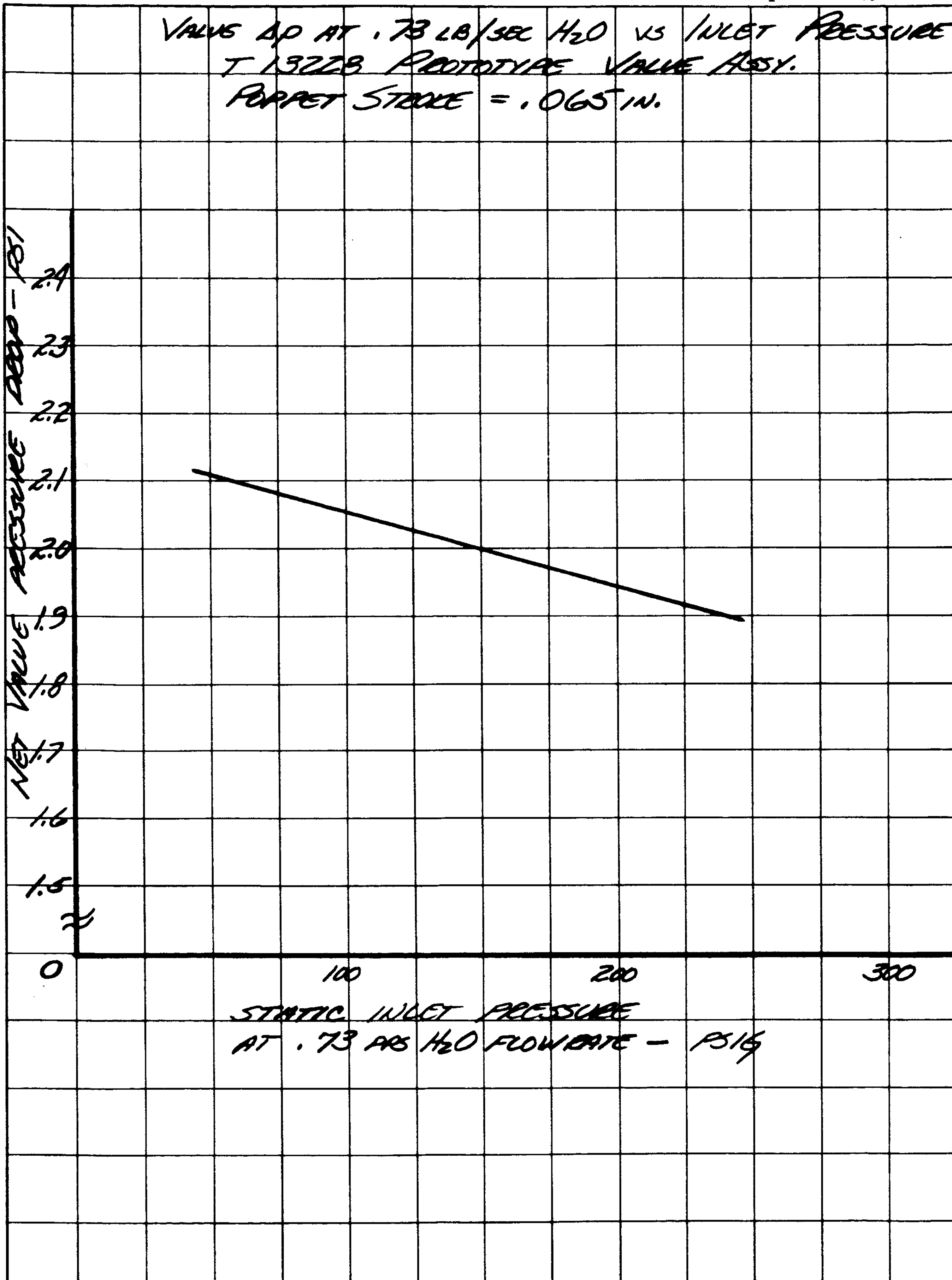
FIGURE 20



NEG. C7001-3

WATER FLOW SETUP - PROTOTYPE TEST
(COPY)20 DEC 65 (U)

FIGURE 21


VALVE ΔP AT .73 LB/SEC H2O VS INLET PRESSURE
T 13228 PROTOTYPE VALVE ASSY.
POPPET STROKE = .065 IN.
NET VALVE PRESSURE DROP - PSI
2.4
2.3
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
0
100
200
300
STATIC INLET PRESSURE
AT .73 PPS H2O FLOWRATE - PSIG

CLOSING RESPONSE
INDUCED VOLTAGE COIL TRACES

T13228 PROTOTYPE VALVE ASSEMBLY (NO ZENER DIODE)

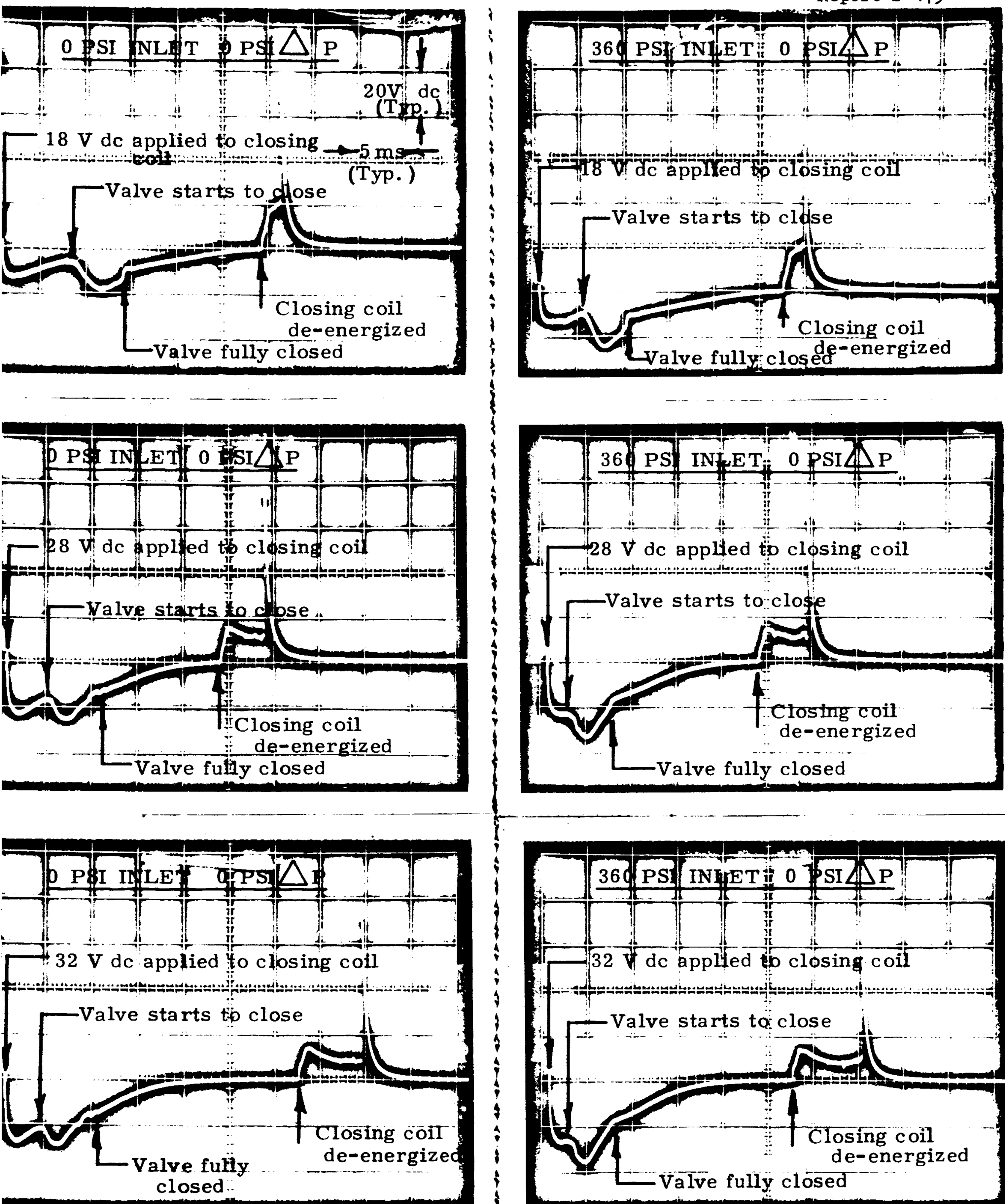


FIGURE 23

OPENING RESPONSE
INDUCED VOLTAGE COIL TRACES

T 13228 PROTOTYPE VALVE ASSEMBLY (NO ZENER DIODE)

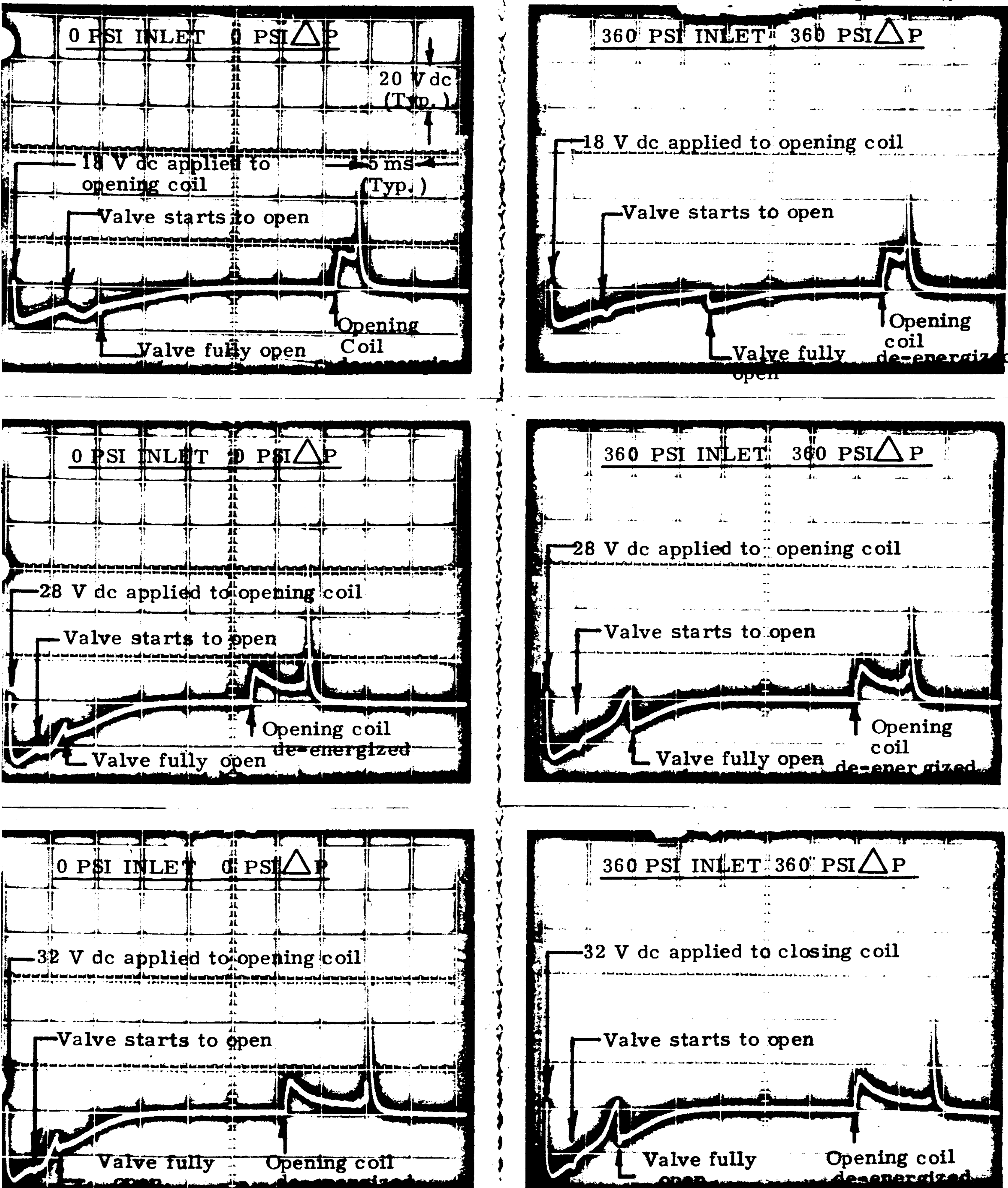
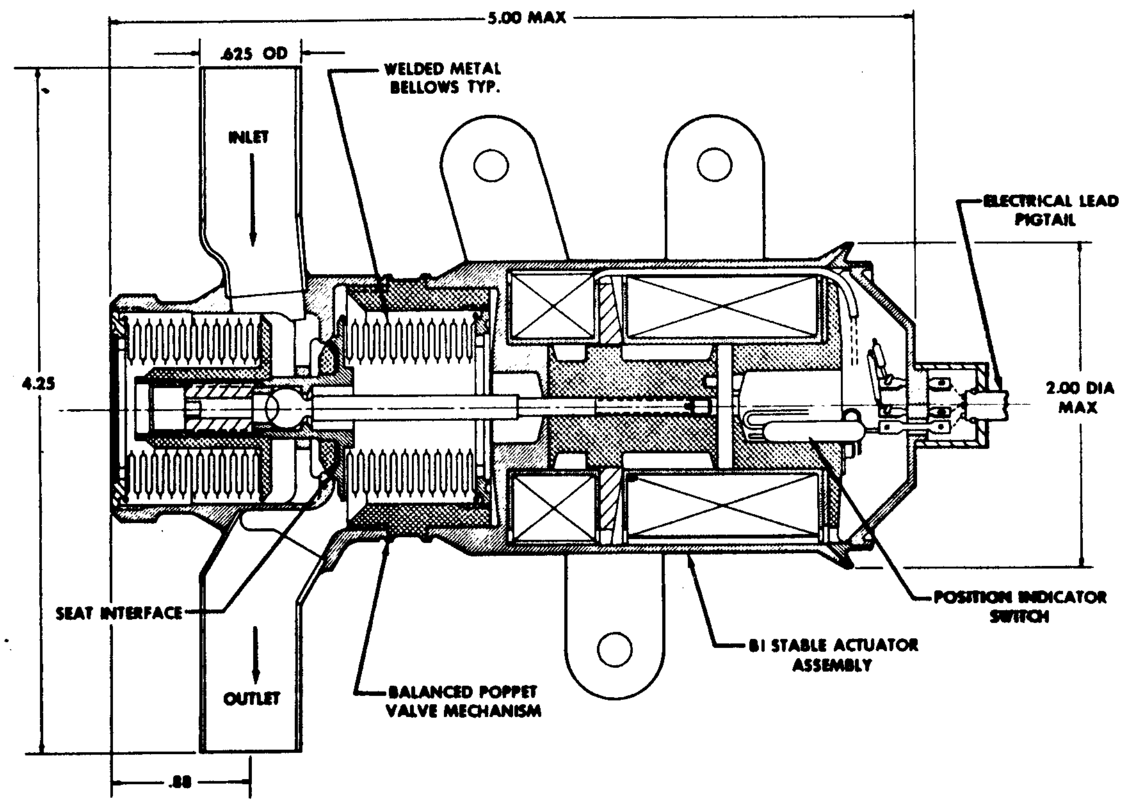


FIGURE 24

MARQUARDT BI-STABLE ACTUATED PROPELLANT SHUT-OFF VALVE

PERFORMANCE CHARACTERISTICS

FLUID SERVICE	N_2O_4, UDMH, MMH, AEROZINE 50
OPERATING PRESSURE	360 PSIG
PROOF PRESSURE	540 PSIG
BURST PRESSURE	720 PSIG
FLOW RATE	.88 PPS N_2O_4 @ $\Delta p = 2.0$ PSI MAX.
VOLTAGE	18 - 32 VDC
CURRENT	2.0 AMP. MAX. @ 30 VDC (WHILE OPENING OR CLOSING ONLY)
OPERATING TEMPERATURE	40 TO 100° F
LEAKAGE: EXT.	5×10^{-6} $\frac{SCC}{SEC}$ He MAX. 0-540 PSIG
INT.	20 $\frac{SCC}{HR.}$ He MAX. 0-360 PSIG
WEIGHT MAX.	1.5 LB. MAX.
WEIGHT	1.4 LB.
WEIGHT, ULTIMATE FLIGHT	1.25 LB.



THE MARQUARDT CORPORATION

V2876-22B
9-30-65

FIGURE 25

APPENDIX A

EXHIBIT "A"

SPECIFICATION
TWO-WAY, LATCHING, D.C. SOLENOID CONTROL VALVE

APPENDIX A

EXHIBIT "A"

Specification

TWO (2) WAY, LATCHING, D. C. SOLENOID CONTROL VALVE

This specification establishes the requirements for the design and fabrication of a flight weight two way, latching, D.C. solenoid control valve, hereafter referred to as the Unit. Two (2) Units shall have a flow rate of 0.88 lb/sec of 70°F propellant (N_2O_4) at a pressure drop of 2 psi maximum. One (1) Unit shall have a flow rate of 0.44 lb/sec of Aerozine-50 at a pressure drop of 2 psi maximum at 70°F. The Units shall conform to the envelope of Figure 2.

Specifications are as follows:

1. Components - Each unit shall consist of the following: (a) valve assembly, (b) latching mechanism for open and closed positions, (c) position indicator switch, and (d) solenoid(s).

2. Materials - Materials must be compatible with monomethylhydrazine (MMH), unsymmetrical dimethylhydrazine (UDMH), nitrogen tetroxide (N_2O_4), mixed oxides of nitrogen, and de-ionized water on a continuous or intermittent basis. A prime design objective shall be a unit compatible with high concentrations of nitric acid.

3. Finishes - Cadmium or zinc plating shall not be used on external surfaces. It is also preferred that no plating be used on any material exposed to propellants for the purpose of making that material compatible.

4. Design and Construction - The Unit shall be designed and constructed in accordance with the requirements of this specification and shall be capable of withstanding the strains, shocks, vibration, and other conditions incident to operational service as specified herein. This shall not be demonstrated by the Contractor.

APPENDIX A (Continued)

5. The Unit should be simple in construction and incorporate the minimum sliding and parts.

6. The Unit shall be designed for easy flushing and incorporate the minimum cavities which would tend to entrap fluid.

7. Lubrication - The Unit shall not require lubrication during storage and service life.

8. Weight - The weight of the Unit shall be kept to a minimum consistent with good design and shall not exceed 1.5 pounds. (Design objective 1.15 pounds.) Minimum weight shall be a prime design factor.

9. Installation - Installation of the Unit into the spacecraft will be by brazing with a water-jacketed induction tool. The Unit shall, therefore, be capable of meeting all functional requirements specified herein after exposure to six thermal cycles consisting of two minutes of 400°F temperature on the port nipples.

10. Electrical Requirements - Unless otherwise specified, all electrical components used in the Unit shall be in accordance with specification MIL-D-9402.

11. Position Indicator Switch - The Unit shall incorporate an electrical switch which shall be open when the valve is open, and which shall provide electrical continuity when the valve is closed. The design shall be such that erroneous readings are highly improbable.

12. The indicator switch shall have a minimum rating of 2.5 watts and contact resistance shall be 50 milliohms maximum. It shall operate from 28 volts - D.C. nominal.

13. Solenoid - The solenoid(s) shall conform to the requirements of specification MIL-S-4040.

14. Pigtails - Three feet long pigtails shall be used and hermetically sealed at the valve body.

APPENDIX A (Continued)

15. Performance - The Unit shall be capable of being used in the propellant system (Figure 1) to interrupt the supply of propellant to the engines in the event of a failure of downstream components, and to provide a method of isolation during ground checkout of the system.

16. Electrical - The Unit shall be capable of withstanding 2 minutes continuous duty operation with 10 minutes between consecutive operations. Operation in excess of 2 minutes continuous duty shall not result in sparking, deterioration of the propellant, or a deterioration of the Unit's burst pressure integrity.

17. Current - Current shall not exceed 2 amperes at 30 volts - D.C.

18. Pull-in Voltage - Solenoid pull-in voltage shall not exceed 18 volts - D.C.

19. Insulation Resistance - 100 megohms minimum at 500 volts - D.C.

20. Dielectric Strength - 1500 volts - A.C. at 60 cps for one minute without electrical breakdown, flashover or current flow in excess of 0.5 milliamperes.

21. Response Time - Under any operating conditions specified herein, the Unit shall be capable of completing the opened-to-closed or the closed-to-opened cycle in 200 milliseconds or less.

22. Temperature - Operating + 40°F to + 100°F; non-operating -65°F to + 160°F

23. External Leakage - There shall be no evidence of external propellant leakage when the valve is cycled and when the valve is maintained in the open or close position at any propellant pressure of 540 psig or less. In addition, when the Unit is pressurized with helium, external leakage shall not exceed 5×10^{-6} standard cc of helium per second at any internal pressure of 540 psig or less.

24. Internal Leakage - With the valve in the closed position and with any propellant pressure of 360 psig or less applied at the Unit inlet, there shall be no evidence of propellant leakage from the outlet port.

APPENDIX A (Continued)

In addition, with helium applied at the inlet port at a pressure of 360 psig or less, leakage at the outlet port shall not exceed 20 standard cc of helium per hour.

25. Operating Pressure - Operating pressure shall be from zero to 360 psig at both the inlet and outlet ports, with the inlet port pressure equal to, or greater than, the outlet port pressure.

26. Proof Pressure - Unit shall be capable of withstanding a pressure of 540 psig without damage or permanent deformation.

27. Burst Pressure - Unit shall be capable of withstanding 750 psig without rupture.

28. Reliable Operating Life - The Unit shall have a reliable operating life of 20,000 cycles.

29. Operating Attitude - The Unit shall be capable of satisfactory performance in all attitudes.

30. Operating Environment - The criteria described in this section represent the environmental conditions and levels to which the equipment will be subjected during the flight mission phases.

(a)	Temperature	Ambient, plus 40 to 80 degrees F. Fluid, plus 40 to plus 80 degrees F.
(b)	Humidity	95 (plus or minus 5) percent including condensation of water.
(c)	Pressure	14.7 psig to 1 x 10^{-6} mm. of mercury for 15 days.
(d)	Acceleration	20 g for 147 seconds.
(e)	Vibration	See Table A-I.

APPENDIX A (Continued)

(f) Acoustics

Octave band cps	Sound Pressures (decibles) (From a sound level reference point of 0.002 Dynes/cm^2)
4.7 to 9.4	145
9.4 to 18.8	153
18.8 to 37.5	150
37.5 to 75	150
75 to 150	147
150 to 300	145
300 to 600	140
600 to 1,200	132
1,200 to 2,400	125
2,400 to 4,800	120
4,800 to 9.600	116
Over-all	158

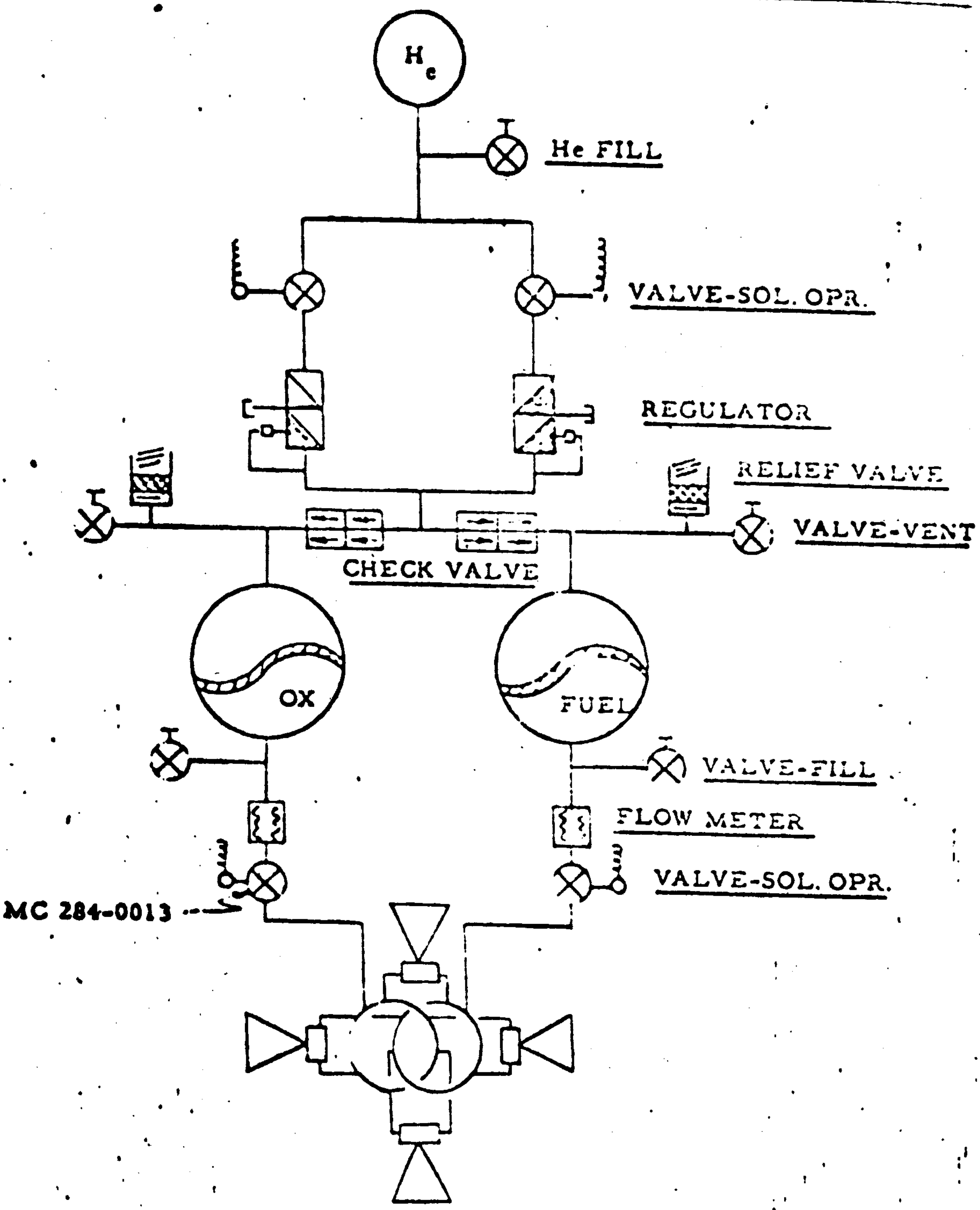
TABLE A-I VIBRATION REQUIREMENTS

Test	Minutes per Axis	Test Method
QUALIFICATION LEVEL	15	RANDOM: Linear Increase from 0.006 g^2/cps at 5/cps to 0.155 g^2/cps at 60 cps. Constant 0.155 g^2/cps from 60 cps to 500 cps Linear decrease from 0.155 g^2/cps at 500 cps to 0.035 g^2/cps at 2000 cps.
	5	Superimposed Sinusoidal Vibration - Linear increase from 0.3 g at 5 cps to 8.0 g at 90 cps. Constant 8.0 g from 90 cps to 300 cps Step increase from 8 to 12 g at 300 cps, constant 12 g from 300 cps to 2000 cps.

NOTE: References to linear are predicated on log-log scale relationships.

APPENDIX A (Continued)

31. Transient inductive voltage caused by either energization or deenergization of the valve's solenoid coils shall not produce a transient voltage "spike" of more than 50 volts and not to exceed 100 ms duration, measure across the leads of each solenoid coil.



SCHEMATIC DIAGRAM
RCS SERVICE MODULE
FIGURE I.

37

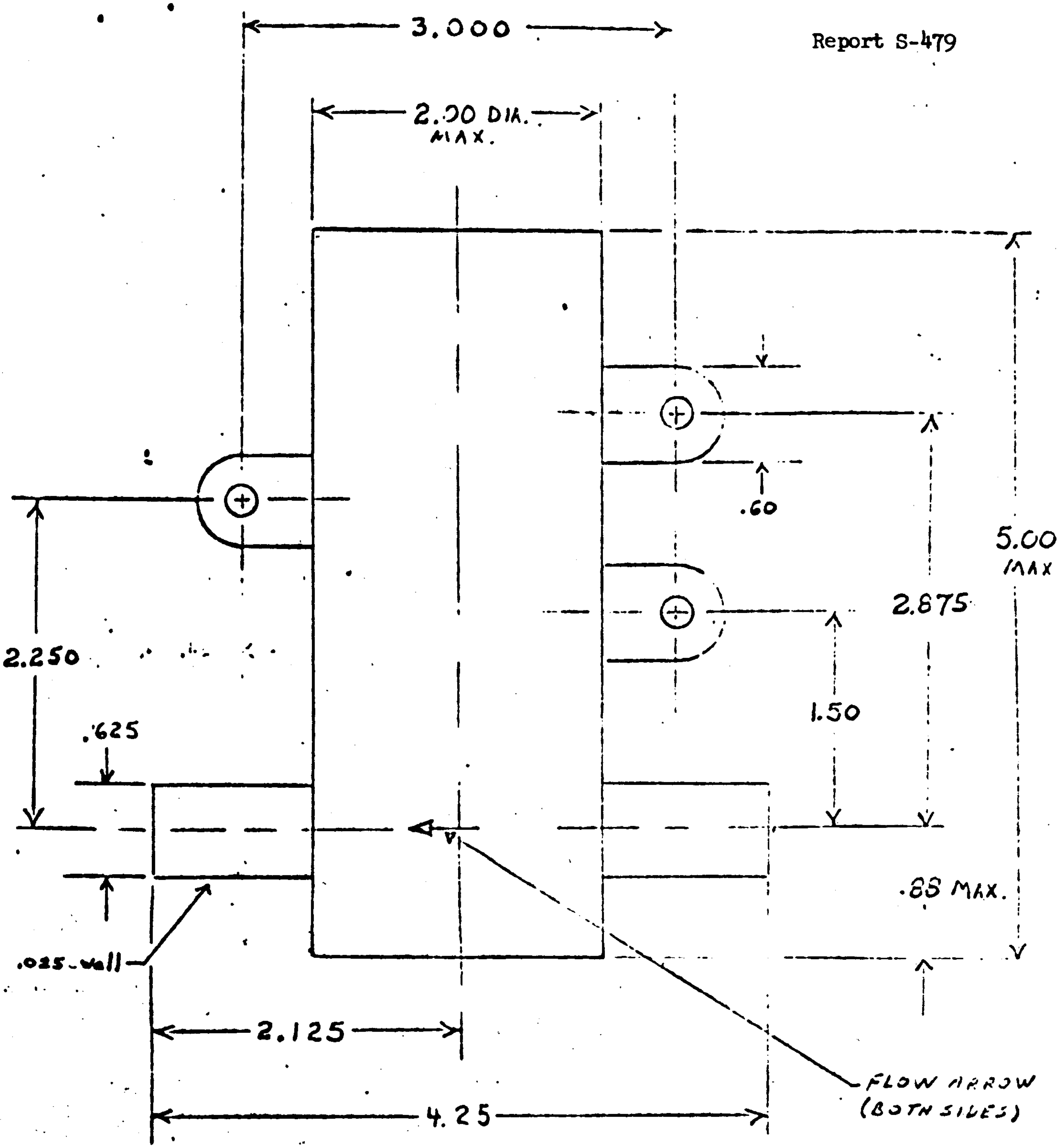


Fig. 2 – VALVE ENVELOPE